Independent Front Suspension on Trucks

Ride comfort analysis and improvements

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TU/e Master's Thesis

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Preface

Over the past year I have been working on my master’s thesis. It started with answering a question coming from the truck market. Answering this question resulted in many more questions, which take most of the time of my master project to answer them. At the end of my master project I have researched and developed some goals to deal with the most important problem.

In the beginning of my master project I had intensive contact with different people form the truck industry, who submitted the question. To answer this question I moved in to the automotive engineering science laboratory which is located within the mechanical engineering facility of the TU/e. Here I had the most important tools like the, the tractor semi-trailer multi-body model to my disposal. In the mean time I remained in contact with the truck market, while answering their questions.

I was working with different people which all had there own view on the problem. During this master project I learned to search for an answer using multi-body theory in a multi-body simulation model. I learned to develop highly sophisticated multi-body models which can give an answer to all kind of problems.

I want to thank all the people who have helped and supported me during this master project. To begin Professor Henk Nijmeijer for his overlook and Igo Besselink for his guidance and in depth technological help, correcting and involving me into everything concerning the master project. Also thanks to the people of truck market which for confidentiality I don’t mention by name, who posed the beginning question. Besides those people I want to thank everyone who has assisted and supported me during my master thesis.

In my personal life I want to thank my family and friends for their support. They helped me to continue during the hard times of my master like writing this report, and in particular my beloved girlfriend for her supporting words.

Pieter Fleuren, June 2009
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Dutch summary

In dit rapport staan de resultaten beschreven van het onderzoek naar onafhankelijke voorwielophanging toegepast op een vrachtwagen trekker-oplegger combinatie. Een bestaand Multi-body model van een trekker-oplegger combinatie is uitgebreid met een onafhankelijke voorwielophanging multi-body blok. Dit modelblok is ontworpen voor het simuleren van verschillende situaties ten behoeve van het onderzoek.

In het eerste deel van het onderdeel wordt de onafhankelijke voorwielophanging vergeleken met de traditionele bladgeveerde starre as voorwielophanging, welke zich door de jaren heen heeft bewezen als een betrouwbare en comfortabele ophanging. De eerste simulatie resultaten zijn teleurstellend. Het verwachte hoge rijcomfort van de onafhankelijke voorwielophanging valt tegen en blijkt na bestuderen van de eerste simulaties zelfs lager te zijn dan de traditionele starre as voorwielophanging.

Na het analyseren van de eerste simulatie resultaten is gezocht naar de factoren die het teleurstellende rijcomfort veroorzaken. Waaruit blijkt dat de torsie stijfheid van een vrachtwagen chassis de grootste oorzaak is voor het lage rijcomfort. In vergelijking met de personenauto, welke gebruik maakt van een stijf zelfdragende carrosserie, heeft een vrachtwagen chassis een lage torsie stijfheid. Dit wetende dat de onafhankelijke voorwielophanging een goede staat heeft betreffende het rijcomfort van een personenauto.

Omdat dit onderzoek niet de intentie heeft het vrachtwagen chassis te verbeter/vernieuwen zijn de invloeden van de meest essentiële onderdelen van de onafhankelijke voorwielophanging onderzocht met betrekking op het rijcomfort. Welke bestaan uit de stijfheid van de luchtvering, de demping van de schokdempers en de onafgeveerde massa. Verandering in veerstijfheid verandert het rijcomfort weinig in vergelijking met verandering van de onafgeveerde massa. Verandering van de demping biedt de meeste mogelijkheid op verbetering van het rijcomfort. Door de juiste keuze en afstelling van de schokdempers kan een goed rijcomfort gerealiseerd worden.

Ten behoeve van rijcomfort verhoging doormiddel van aanpassing van de demping van de onafhankelijke voorwielophanging is gezocht naar een alternatieve schokdemper. De Frequentie Selectie Schokdemper afgekort met FSD heeft, zo blijkt uit een uitvoerige studie naar de verschillende alternatieve schokdempers, de meeste potenties om het rijcomfort te verbeteren. Van de FSD is een multi-body model gemaakt welke verwisseld kan worden met de in de origineel trekker-oplegger model gebruikten schokdempers. Na simulatie van de onafhankelijke voorwielophanging met optimale afgestelde FSD demping is een rijcomfort verbetering van 30% zichtbaar ten opzicht van de eerder onderzochte onafhankelijke voorwielophanging met originele schokdempers. Na simulatie van FSD demping in combinatie met de starre as voorwielophanging is ook hierbij een 8% rijcomfort verbetering zichtbaar.
Na uitvoering onderzoek is gebleken dat gebruik van een onafhankelijke voorwielophanging onder een vrachtwagen chassis niet tot een verbetering van het rijcomfort leid. Alleen na de juiste keuze van schokdempers met optimale set-up of een alternatief torsiestijf geraamd chassis kan een rijcomfort worden bereikt welke groter/vergelijkbaar is met die van de traditionele starre as voorwielophanging.
English summary

This report describes the results of analysis of an Independent Front Suspension used in a tractor semi-trailer truck configuration. After designing an independent front suspension multi-body model which is build into an existing tractor-trailer truck model different simulations are executed.

First the IFS is compared with the usually used and many life-times proven rigid beam suspension. The results are disappointing; the expected comfort improvement when using the IFS stays out. The result shows that even the comfort of the IFS was deteriorated compared to the old proven rigid beam suspension.

After this disappointing discovery a search is made for the most responsible factor of the ride comfort. After analysis of different simulation results it appears that the low torsion stiffness of the truck's chassis is responsible for the disappointing ride comfort. Looking to the passenger car industry we see favourable ride comfort effects using IFS in combination with a stiff integral-frame construction in combination with IFS has.

Because this investigation is not intended to improve or adapt the truck chassis, the most essential part of the IFS, namely the spring, dampers and unsprung mass must be analysed, to see what the influences are on the ride comfort. Variation in spring stiffness will not influence the ride comfort a lot, while lowering the unsprung mass will. Most effect to the ride comfort has the choice of the shock absorbers and their settings. They must be improved to create a better ride comfort for a truck equipped with IFS.

Next step in the IFS investigation is looking for alternative shock absorbers. After reviewing all kind of different damper types one is chosen as the one with the biggest potential to improve the IFS ride comfort, namely the Frequency Selective Damper called FSD. This type of damper is build into the IFS multi-body model to simulate the tractor-trailer ride comfort with FSD dampers. After tuning the most important FSD parameters for heavy truck use significant ride comfort improvement is made. Improvements of 30% for the IFS configuration and also a less high but nice 8% for the rigid beam suspension are possible.

When using an independent front suspension in a heavy truck configuration it can reach the same ride comfort as using the proven rigid beam suspension, by choosing the right shock absorbers with fine tuned settings. After all the IFS will have no big ride comfort advantages comparing to existing rigid beam suspensions when using the usual window-frame truck chassis.
1 Introduction

Ride comfort is one of the most important qualities of a heavy commercial vehicle like a truck. All kind of suspensions are developed to improve the ride comfort. Partially the front suspension is decisive for the ride comfort, because above the cabin with the driver’s seat is installed.

A truck is never equipped with an independent front suspension also called “IFS”, which is often found in passenger cars. In the passenger car market the independent suspension is a proven concept. In the truck market they apply different rules, they often improve their product. Replace them for a new innovation design is very expensive and risky in an environment with a lot of competition.

But exactly those innovations can make the different on the market with respect to the competition, so your truck will be a best-seller. So can a truck equipped with IFS give a truck manufacture the leading position on the market regarding to comfort and drivability? During some first test drives with IFS on a truck the expected ride comfort, like in the passenger car industry, stays out. Looking to the fact a truck is build up around a frame and a passenger car uses a frameless construction, an important question remains;

What is happening to the ride comfort using an independent suspension on a commercial vehicle with a flexible chassis?

This question started the investigation of independent front suspension on trucks. In the first part of this report chapters 3, 4 and 5 this question is answered using a multi-body model of a tractor semi-trailer configuration equipped with an independent front suspension. Foregoing different literature was studied to have the necessarily information about the front suspension kinematics, truck front suspension types, independent suspension on commercial vehicles and independent suspensions for passenger cars.

The used design of the IFS on trucks will explained in chapter 3. This is a simplified design of the nowadays complex constructions mounted in passenger cars. Chapter 4 describes how this design is build into a working sim-mechanics multi-body model, which is capable to simulate a complex tractor semi-trailer model and derive the necessary equations which give the required results to answer the main question. The simulation results of the IFS are compared with reference semi-trailer configuration, which is equipped with the original rigid beam suspension with leafsprings. The main question with respect to the flexible chassis is answered with those simulation results.
At the end of chapter 5 an imperative sensitivity analysis is worked out. What are the influences of the three most important parameters on the in the truck market so important ride comfort? To get the right answers the tractor semi-trailer is simulated with variations in spring stiffness, damping and unsprung mass. Investigating those parameters will result in better problem understanding and will give a direction to look at for possible solutions.

In the last part, chapter 7, some technology developments like position sensitive damping (PSD), acceleration sensitive damping (ASD), frequency selective damping (FSD) and continuous camping control (CDC) are introduced considering the most sensitive parameter the damping. The best alternative is built in the tractor semi-trailer multi-body model. Simulation of this innovation gives some improving results. A short parameter study explains the effects this technology development more precisely.

In chapter 7 the conclusions is drawn and some recommendations are given.
2 Literature survey

Suspension is the term given to the system of springs, shock absorbers and linkages that connects wheels to a vehicle. Suspension systems serve a dual purpose as shown in Table 2-1; contributing to the car’s handling and braking for good active safety and driving pleasure, and keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations. These goals are generally at odds, so the tuning of suspensions involves finding the right compromise. The suspension also protects the vehicle itself and any cargo or luggage from damage and wear.

<table>
<thead>
<tr>
<th>Principle</th>
<th>Definition</th>
<th>Goal</th>
<th>Solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Road Isolation</td>
<td>The vehicle’s ability to absorb or isolate road shock from the passenger compartment</td>
<td>Allow the vehicle body to ride undisturbed while traveling over rough roads.</td>
<td>Absorb energy from road bumps and dissipate it without causing undue oscillation in the vehicle.</td>
</tr>
<tr>
<td>Road Holding</td>
<td>The degree to which a car maintains contact with the road surface in various types of directional changes and in a straight line</td>
<td>Keep the tires in contact with the ground, because it is the friction between the tires and the road that affects a vehicle’s ability to steer, brake and accelerate.</td>
<td>Minimize the transfer of vehicle weight from side to side and front to back, as this transfer of weight reduces the tire’s grip on the road.</td>
</tr>
<tr>
<td>Cornering</td>
<td>The ability of a vehicle to travel a curved path</td>
<td>Minimize body roll, which occurs as centrifugal force pushes outward on a car’s center of gravity while cornering, raising one side of the vehicle and lowering the opposite side.</td>
<td>Transfer the weight of the car during cornering from the left to right or reversed.</td>
</tr>
</tbody>
</table>

Table 2-1: Three main functions of the suspension; Road Isolation, Road Holding and Cornering

It is well known that the excessive levels of vibration in commercial vehicles negatively affect driver comfort, cargo safety and road condition. The current challenge in the field of suspension design for heavy vehicles is to optimize the suspension dynamic parameters to improve such requirements. One possible solution is an independent front suspension abbreviated as IFS.

In this chapter first the suspension kinematics like camber, caster, toe and KPI are explained. These are fundamental characteristics to understand a vehicle suspension. Next an overview of different front suspension types of commercial vehicles is summed. In paragraph 2.3 the only available paper concerning independent front suspension on trucks is discussed. Because there is almost none literature available concerning the IFS on trucks, also independent front suspension on passenger cars is discussed. In particular the history of the independent suspension on cars is interesting. The reason for using an independent front suspension on passenger cars can help find a motivation for introducing on a commercial vehicle.
2.1. Front suspension kinematics

The stability and effective handling of a vehicle depends upon the designers' selection of the optimum steering and suspension geometry which particularly includes the suspension kinematics. The suspension kinematics can characterize by different values such as camber, caster and kingpin inclination. The most important suspension kinematics are explained in this paragraph.

**Camber**

The camber angle is defined as the angle between the tilted wheel plane and the vertical plane (see Figure 2-1). The camber is positive if the wheel leans outward at the top relative to the vehicle, or negative if it leans inward as shown in Figure 2-2. The static camber angle is used in conjunction with the camber change rate of the suspension to position the tire at the optimum camber angle while turning. A cambered rolling pneumatic-tired wheel produces a lateral force in the direction of the tilt. Camber also affects the pneumatic trail. The aligning torque due to camber is generally quite small because of the symmetry of the tire print distortion.
Camber change rate

The camber change rate is the amount of camber change during suspension travel. This rather can be bump or body roll and depend on the suspension type (see Figure 2-3).

Scrub

Tire scrub is the lateral motion relative to the ground that results from vertical motion of the wheels. The amount of scrub is dependent on the suspension type.

Kingpin inclination KPI

KPI is the angle of the rotation axle of the front wheels in the front view (y-z) plane (see Figure 2-4 and Figure 2-1). The more positive KPI the more the truck will be raised while steering. The effect of raising the truck is to aid centering of the steering at low speed. Another effect of KPI is changing the camber angle when turning the wheels. When a wheel is steered, it will lean out at the top, toward positive camber, if the kingpin is inclined in the normal direction (toward the center of the car at the upper end).

Spindle length

The Spindle length is the distance between kingpin axle and the axle parallel to the kingpin axle which leads through the wheel center (see Figure 2-4). With a positive spindle length the truck will be raised upwards as the wheels are steered, for a positive KPI a longer positive spindle length will increase the amount of lift with steer. This effect is symmetric side to side if there is no caster angle. The effect is self-centering at low speeds.

Figure 2-4: Kingpin geometry (ref. [1])
**Scrub radius**

Scrub radius or kingpin offset is the lateral distance in the front view y-z plane between wheel plane of symmetry and the intersection of the kingpin axis with the road (see Figure 2-4). When the scrub radius isn’t zero braking and driving forces can introduce steer torque when left and right are not the same, which for example occurs during cornering. Also when the forces are different at the left and right wheel there will be a net steering torque felt by the driver.

**Caster angle**

Caster is the angle of the rotation axle (kingpin axis) of the wheels in the side view (x-z) plane (see Figure 2-4 and Figure 2-1). Caster angle, like KPI causes the wheel to rise and fall with steer. Unlike KPI, the effect is opposite from side to side. The effect of left steer is to roll the car to the right, causing diagonal weight shift, which can correct the over- and understeer characteristics. Also the caster angle effects camber due to steering but, unlike KPI, the effect is favorable. With positive caster angle the outside wheel will camber in a negative direction while the inside wheel cambers in a positive direction, again leaning into the turn.

**Caster change rate**

Caster change rate is the amount of caster change during suspension travel. There is very little reason to intentionally have caster change. Because mostly the steering arm can not be placed on wheel center height, it will describe a different circular path during caster change comparing to the rigid axle beam or upright, this caused bump-steer.
8.4 Ride comfort weighting functions

Mechanical trail

The mechanical trail or caster offset is the distance in the side view x-z plane between tire contact patch and the intersection of the kingpin axle with the road (see Figure 2-4 and Figure 2-5). The mechanical trail is obtained by the caster angle and the kingpin offset. The kingpin offset is the distance between the kingpin and the wheel center. With positive mechanical trail the tire contact patch is behind the kingpin-axis in the side view. More trail means that the tire side force has a larger moment arm to act on the kingpin axis. This produces a higher steering torque and increased self-centering effects.

Pneumatic trail

The pneumatic trail is the distance from the force-aft center of the tire road contact point to the center of action of the lateral force (see Figure 2-5). This creates a tire aligning torque which is the lateral force times the pneumatic trail. The pneumatic trail has same effects as the mechanical trail however, this tire effect is nonlinear with lateral force and affect steering torque and driver feel.

Ackermann

As the front wheels of a vehicle are steered away from the straight-ahead position, the design of the steering linkage will determine if the wheels stay parallel or if one wheel steers more than the other the so-called Ackermann steering geometry. There are three steering geometry possibilities; Ackerman, Parallel and Reverse Ackermann (see Figure 2-6). For low lateral acceleration usage (street cars and trucks) it is common to use Ackermann geometry, this ensures that all the wheels roll freely with no slip angles because the wheels steered to track a common turn center. For high accelerations the tires operate at significant slip angles and the loads on the inside track are much less than on the outside track. Less slip angle is required at lighter loads to reach the peak of the cornering force curve; so for this case a steering geometry in direction towards reverse Ackermann is recommendable.

Figure 2-6: Ackermann steering geometry (ref. [1])
Toe-in

The static toe-in angle is the angle that results in a standing vehicle, between the vehicle centre plane in the longitudinal direction and the line intersecting the center plane of the left or right wheel with the road plane. It is positive when the front part of the wheel is turned towards the vehicle longitudinal centre plane and negative (toe-out) when it is turned away. The amount depends on other suspension parameters.

Ride and Roll steer

Ride and Roll is the result from wheel ride (bump) motion and body roll (and pitch) motion. They are undesirable because it causes the car turning on a path that the driver did not select. They are a function of the suspension and steering geometry. If the tie rod is not aimed at the instant axis (IC in Figure 2-7) then steer will occur with ride because the steering and suspension are moving about different centers. Also if the tie rod is not the correct length for its location then it will not continue to point at the instant axis when the suspension is traveling.

Roll centers and axis

The roll centre (RC in Figure 2-7) of a suspension system refers to that centre relative to the ground about which the body will instantaneously rotate. The actual position of the roll centre varies with the geometry of the suspension and the angle of roll. The roll axis (C-C in Figure 2-8) is the line joining the roll centers of the front and the rear suspension. Roll centre height for the front and rear suspension will be quite different; usually the front suspension has a lower roll centre than that at the rear, causing the roll axis to slope down towards the front of the vehicle.

Figure 2-7: Roll center RC with the instant centers IC (ref. [2])
Figure 2-8: Roll centers with roll axis CC (ref. [2])

Anti features

The anti-effect in suspension is a term that actually describes the longitudinal to vertical force coupling between the sprung and unsprung masses. It results purely from the angle or slope of the side view swing arm.
The anti-effects:

- Anti-dive (Figure 2-9) geometry in front suspensions reduces the bump deflection under forward braking.
- Anti-lift in front suspensions only occurs with front-wheel drive and it reduces the suspension droop deflection under forward acceleration.
- Anti-rise (Figure 2-9) in rear suspension reduces droop travel in forward braking.
- Anti-squat in rear suspensions reduces the bump travel during forward acceleration on rear wheel drive cars only.

The amount of those anti-effects is depending on the torque reaction (created by acceleration or braking) taken by chassis (inboard braking and independent suspension acceleration) or control arms (outboard braking and solid axle acceleration).

2.2. Truck front suspension types

Different front suspension types exist on the truck market. The most common type is the rigid axle with two longitudinally placed leafsprings. The rigid axle front suspension is an old suspension type which is still in use. Nowadays the commercial market asks for more and better properties of the truck, which results in new front suspension designs. One of the main changes is the introduction of the airspring. The rigid axle airspring suspension provides a better ride comfort, which is very important in the truck industry. The airspring frontaxle is a successful front suspension type, but because of its good performance and simplicity the leafspring suspension still is the most frequently occurring design. Because the truck customers demands to even better ride comfort and suspension requirements the truck manufacturers are investigating new front suspension designs such as an independent front suspension. In this paragraph the different suspension types will shown.
Rigid axle suspension

Leafspring

The rigid axle leafspring suspension exists of a rigid beam, which is connected to two longitudinally placed leafsprings (see Figure 2-10). In the most common configuration the leafsprings are installed under the window chassis' two longitudinal runners. The dampers are mounted on the outside of the chassis. The leafsprings have two functions; making vertical wheel travel possible and react against the longitudinal and lateral axle forces. To create enough body roll stiffness an anti-roll bar is installed.

![Figure 2-10: Rigid axle leafspring suspension](Source: DAF XF95 7500kg Leafspring Frontaxle)

![Figure 2-11: Rigid axle simply airspring suspension](Source: DAF XF95 7500kg Airspring Frontaxle)

Airspring

In the simplest rigid axle airspring configuration (Figure 2-11), the hybrid airspring suspension, the original leafsprings are replaced by less stiff ones. Between the rigid axle and the longitudinal runners two airsprings are placed. In this design the leafsprings are less stiff and no longer able to control the brake moment, so it is necessary to place an extra reaction arm in longitudinal direction. Also like the original leafspring suspension there are two dampers and an anti-roll bar installed.

![Figure 2-12: Frontaxle with stabilenker (source: Mercedes Benz Actros)](source: Mercedes Benz Actros)

A more common solution is the five rods airspring suspension (Figure 2-13). In this design five reaction arms are responsible for the location of the axle. Four 2x2 parallel mounted arms take care of the longitudinal forces and a panhard arm is installed behind the axle to take care of the lateral forces. Like an original leafspring suspension there are two dampers placed on the
outside of the longitudinal runners and an anti-roll bar prevents excessive body roll. This design also can be equipped with a stabilenker (Figure 2-12). In the stabilenker configuration the lower two longitudinal reaction arms are intergraded in an anti-roll bar which is called the stabilenker. This saves parts and shortens the assembly time.

Another development is the trailing arm airspring suspension (Figure 2-14). In this design the anti-roll bar is integrated in the two longitudinal reaction arms which are fixed to the rigid axle, so the rigid axle act like the conventionally torsion beam rearaxle as seen on many passenger cars. Also the airspings and dampers are integrated, and placed in two suspension struts. A panhard arm takes care of the lateral forces.

**Independent Suspension**

A newer less common suspension design in the heavy commercial truck market is the Independent front suspension (Figure 2-17). In this suspension the rigid axle is replaced by two A-arms with uprights. This independent double wishbone suspension is more common in the automobile industry. In this design the left and right tires are no longer coupled to each other, which means they can travel independently. There is no mutual wheel influence, driving (Figure 2-15). One of the disadvantage is track alternation (Figure 2-16), which is caused by the camber change during wheel travel. When track alternation occurs the tires move in lateral opposite direction which increases the tire wear.
The Lower A-arms are mounted to the longitudinal runners using a sub frame. The upper A-arms are connected to the longitudinal runners directly. To the other side of the upright the upright or fusees are mounted. The difference between uprights and fusees is the wheel turning possibility. The uprights can turn around in the A-arm ball joints and so provides the wheels turning. When using fusees the upright aren’t mounted in ball joints but in revolution joints which can’t rotate, so a separated fusee is necessary to turn the wheels, see Figure 2-17.

The airsprings and dampers can be integrated like a trailing arm airspring suspension. Mostly the airsprings and dampers are mounted to the uprights; this is only possible if the uprights can not rotate so these are equipped with a separate fusee (Figure 2-17). The airspring and damper displacement is in this situation the same as the wheel travel. In the other situation the airsprings and/or dampers are mounted to the A-arms. Rotating uprights are possible now, while spring- and damper-rates during wheel travel determines the spring and damper characteristics. The steering geometry design has to taken into account as well, so the bump and roll steer is minimized. This can be realized by fitting the length and position of the tie (steer) rods to the A-arms, so there is no wheel turning when suspension travel occurs. Appendix A shows some other examples of independent front suspension for heavy vehicles.
2.3. Independent suspension on commercial vehicles

This paragraph will be present the only SAE technical paper (ref. [5]), which gives critical detailed information about the IFS on trucks. In this paper R. Bramberger and Dr. W. Eichlseder of the engineering center of Steyr describe some concepts of independent suspensions. First the features and peculiarities of independent suspension on commercial vehicles are summarized. Second the advantages and disadvantages of the different independent suspension types are shown and in the third paragraph the design considerations of the double wishbone suspension is described. At the end a conclusion is given which summarizes the most important points of this SAE paper.

The fundamental targets were: Constant track and camber under all drive and load conditions (like rigid axle design) due to:
- minimized tire wear
- minimized rolling resistance
- minimized fuel consumption for economic operation of the vehicle

Features and peculiarity

- The wheels are independent mounted by links to the frame. With only 1DOF the wheels are guided exactly. The required space in the wheel housing is smaller compared to rigid axles.
- The desired disconnection of masses is possible so that no tramping behavior can occur. The tuning of damping factors is easier and the damping factor can be made smaller what affects the ride comfort and the handling positively, especially with reduced unsprung masses.
- Compared to the rigid axle the roll center height is lower. So the camber and track deviation in unilateral bounce of the vehicle is reduced. This also reduces the axle tramp and the steering vibrations.
- When the vehicles independent suspension travels parallel a deviation of track and camber will occur, which can cause instability in vehicle handling behavior especially on roads with low friction, the vehicle undesirable moves in lateral direction.
- If the body inclines by the roll angle during cornering, the outer independently suspended wheel takes a positive camber and the inner wheel takes on a negative camber. The ability of the tires to transfer the lateral forces decreases causing a greater slip angle.
- The main advantage of the double wishbone suspension is its kinematic possibilities. The position of the suspension control arms relative to one another can determine both the height of the body roll center and the pitch pole. Moreover, the different lengths can influence the angle movements of the compressing (jouncing) and rebounding wheels, i.e. the change of camber and also the track width change. With shorter upper control arm the bouncing wheel goes into negative camber and the rebounding into positive. This counteracts the change of camber caused by the roll of the body.
Advantages and disadvantages of different suspension types

McPherson struts and strut dampers:

Advantages:
- all parts providing the suspension can be combined into one assembly

Disadvantage:
- less favorable kinematic characteristics
- the friction between piston rod and guide impairs the springing effect
- much space needed above wheel for spring and strut (not available on medium duty truck and on bus vehicles)
- connection between upper strut mount and frame unstiff and expensive
- clamping of strut to steering knuckle very difficult for the axle loads of truck
- expensive spare part

Figure 2-18: McPherson suspension strut (ref. [6])

Double wishbone suspension:

Advantages:
- good kinematic possibilities
- good ride and handling comfort
- small space needed in vertical direction in wheel housing

Disadvantage:
- more space needed besides frame in lateral direction
- king pin inclination high (minimum scrub radius and disturbance distance limited)
- less favorable ratio of spring rate and damper rate

Figure 2-19: Double wishbone suspension (ref. [6])

Multi link suspension:

Advantages:
- good kinematic possibilities
- good ride and handling comfort
- small space needed in vertical direction in wheel housing

Disadvantage:
- much space needed beside frame in lateral direction (especially for control rod)
- higher number of ball joints, bearings and links
- increased unsprung masses
- expensive

Figure 2-20: Multi-link suspension (ref. [6])
The advantage of good kinematic possibilities (small changes of camber, toe-in and track), low unsprung mass at moderate costs (+5-10% compared to Rigid Axle Suspension) is a reason to select the Double Wishbone System as concept for an independent front suspension on a truck.

**The Double Wishbone Design**

**Axle Geometry**

The primary goal is to realize minimum scrub radius in combination with well balanced king pin inclination and nominal wheel camber. For the scrub radius the minimum possible value is realized according available space (ball joint size, brake, etc.). Because of the Double Wishbone design the optimum value of the rigid axle could not fully be achieved. To succeed in the economic operation - the minimized changes of the referenced geometric properties under all wheel travel positions have to be defined as the key requirement. Under slip angle conditions, which will occur in the different wheel deflection positions by the kinematic changes, the rolling resistance is increased what leads to high fuel consumption and tire wear out. To define the position of the pitch pole (longitudinal instant center), roll axis, caster angle change and wheel base change have to be considered. Theoretically (for 100% anti-dive compensation) the pitch pole has to be in the CG of the vehicle. To achieve this, a front arrangement of the control arms has to be chosen were the pitch pole is located very high above road: Such an arrangement causes considerable caster angle change and forward wheel movement during wheel jounce. This is not possible because the impact forces of a road obstacle would be increased too much.

**Wheel travel**

Realization of increased wheel travel in order to
- maximize ride comfort
- minimize dynamic wheel loads
- minimize peak shock and reaction forces

To gain the maximum wheel travel in conjunction with high steering angle the angular capability of ball joints and rubber bushes has to be investigated.

**Steering Geometry**

Lay-out has to meet the kinematic requirements as there are:
- steering deviation from Ackermann characteristics
- improvement of steering ratio around the straight ahead position
- symmetrical ratios LH to RH also with respect to loads in steering linkage

**Compliance**

The primary goal is minimum compliance. This leads to minimum coupling between wheel track and steering motion. The minimum compliance can be reached by stiff metal/rubber joints and steering linkage parts. The high stiffness of these parts provides also a positive influence to the durability and lifetime.
**Controllability**

Due to the self-steering behavior, slightly understeer must select in order to provide lateral driving (handling) safety. Regarding to braking controllability this requires avoiding possible toe-out by means of defined pre-set toe in. Additionally, the concept of minimum compliance is considered to be an important condition to suppress shimmy tendencies.

**Ride Comfort:**

The following design features provide improved Ride Comfort:

- increase of the wheel travel in jounce direction
- reduced friction in the suspension system avoids shock transmission also under small excitations
- reduction of the unsprung mass provides decoupling between wheel and chassis
- It should be mentioned that the axle layout is one of the issues which influences the ride comfort.

**General Items:**

Directional stability features as understeer and oversteer are global characteristics of the vehicle. The dual tires at the rear axle will force the vehicle to understeer, although if the suspension path of the rear spring would have a neutral characteristic.

**SAE paper conclusions**

An Independent Front Suspension gives a lot of improvements like good kinematics possibilities, good ride and handling properties, less installation space required and a lower unsprung mass. But there are also some disadvantages like track deviation during vertical wheel traveling and costs. Further on in this report the described expectations with respect to the ride comfort will be extensively tested using a multi-body model of the double wishbone suspension compared to the normal rigid axle leafspring suspension.

### 2.4. Independent suspension for passenger cars

There is only one relevant paper about independent suspension on trucks which was presented in the previous paragraph. Because it is well-know the truck industry generally lags behind the passenger car industry the independent suspensions of passenger cars are reviewed, to get more information about the independent suspension. It is useful to analyze the history of the independent front suspension on cars. The motivation to use new technologies in the truck business is often the same as in the car business. In the passenger car industry the independent suspension is the term used to describe any arrangement by which the wheels are connected to the chassis in a manner such that the rise and fall of one wheel has no direct effect on the others. Although widely used only since the Second World War, its first application was at least as early as 1878.
Rigid front axle: Reasons for decline in use

Looking to the disadvantages of a rigid front axle of a passenger car can be useful to get arguments to use an independent front suspension. The rigid front axle on passenger car is effectively non-existent these days. There are a number of reasons for this.

The main ones according to Donald Bastow (ref. [7]) are:

1. Single wheel bumps are responsible for gyroscopic torques from both front wheels.
2. Single wheel bumps produce front end steering effects from both front wheels because of the sideways deflection of the contact patches in relation to the car due to the high roll centre.
3. Front axles are liable to tramp.
4. Front brake torque implies considerable caster angle change unless a parallel linkage is provided to control it.
5. The narrow spring track implied by half elliptic springs and front wheel lock clearance imply low front anti-roll resistance and a consequential oversteer effect unless an anti-roll bar is provided.
6. The limitations 3, 4 and 5 make it in practice impossible to get the front springs soft enough to obtain a pitch-free ride.
7. Better utilization for passenger accommodation of the area occupied by the car on the road makes engine and front axle want to occupy the same space.

Some of these limitations can be overcome by a variety of complications. Even then, the result is less satisfactory than an independent front suspension. All these points are also valid for commercial vehicles, even point 7, although in another way. For European trucks, the one with cab over engine design, the still bigger engine needs more installation space which there is not using a rigid front axle.

Independent suspension

As a result of these limitations, American cars in 1935 generally adopted independent front suspension. Poor roads on the European continent encouraged Ford and General Motors to transfer this US technology across the Atlantic to their subsidiaries in Germany and Britain. Some European companies such as Lancia and Morgan had anticipated the Americans moves and quickly all manufacturers adopted independent front wheel suspension even for commercial vans. Over the years many types of independent front suspension have been tried. Many of them have been discarded for a variety of reasons, with only two basic concepts, the double wishbone and the McPherson strut, finding widespread success in many varying forms.

The benefits derived from the adoption of independent front suspension are according to T.K. Garrett, K. Newton and W. Steeds (ref. [8]) described as follows:

1. Because of the vertical travel of the wheels, gyroscopic kicks in the steering system are obviated – with a beam axle, these reactions, which tend to initiate wheel shimmy, are inevitable if only one wheel rises over a bump.
2. Steering effects due to lateral movements of the tyre/road contact patch, as the wheel rises and falls are obviated.
3. Variations in castor angle due to this windup are obviated.
4. A coil spring can be more easily accommodated close to a wheel that has to be steered than a semi-elliptic spring can be.
5. The roll centre can be lower and the spring base wider, so the roll resistance is higher. A secondary effect of the inherently high roll resistance is that softer springing can be used which reduces the tendency to pitch.
6. The engine can be positioned further forward, since it does not have to clear a beam axle – this leaves more space for the occupants of the car.
7. Unsprung mass is lighter, and therefore the ride quality improved.

Following Donald Bastow (ref. [7]) the geometry of any independent suspension should be prepared to meet certain requirements:

1. It must allow the roll centre height to be arranged at a desired level.
2. It must allow cross steering connections to be made to each wheel knuckle that induce minimal variation of toe settings with vertical wheel movement.
3. It should allow anti-brake dive geometry to be incorporated if this is required.
4. It must allow coil springs, torsion bars or other springing means to give a desired load-deflection diagram.
5. It should be possible to incorporate telescopic dampers.
6. It should allow an anti-roll bar to be added if necessary.
7. It should not prevent some overlap of the engine with the front suspension to allow the driver’s pedals to be installed with minimum offset from the steering centerline position.
8. It must withstand all the forces imposed on it by braking, acceleration and cornering with the ability to isolate the body structure from noise, vibration and harshness.
9. It should restrict the inertia, gyroscopic or other forces produced by vertical wheel travel.
10. It should create the minimum amount of friction associated with its vertical movement.
11. It should operate without hysteresis throughout its full range of travel and reach its limits progressively.

**Double wishbone**

Donald Bastow (ref. [7]) also gives a view of the initial design and tells why the automobile manufactures adopted the independent double wishbone front suspension:

There are several ways in which pairs of parallel arms can be arranged to control suspension geometry. There can be two upper an lower trailing arms on each side, used with transverse torsion bars as on the classic VW Beetle layout, transverse tubular symmetrically arranged A-frame links or angled derivatives and hybrids. Parallel motion from equal-length links maintains wheel camber angles to be generated by body roll to the detriment of cornering power. The way to control camber better is to fit shorter upper links than lower ones, so that the top of the wheel moves in as the suspension is displaced from the mean travel position. In many applications
dating from the time when lever-arm hydraulic dampers were popular, the short upper link was formed by the damper-actuating lever.

If a substantial front sub-frame forms part of the inherent vehicle design, double wishbone can be designed to attenuate road noise better. For precision control of the system, close manufacturing tolerances are vital for any lay out and it is usual to machine the attachments after sub-frame assembly. On the Jaguar XJ6 (see Figure 2-21), for example, the dampers are attached directly to the body structure to ensure that the damping forces on the body are not depleted by the insulation of the sub-frame mounts.

The first design to depart from this was the post World War Two 2.6 liter Lagonda. It had become fairly general practice for the axes to be splayed out toward the front of the car; if the lower lever was longer, either to get or approach straight line motion of the tire road contact point up and down or to be able to use a two-piece cross steering tube with central idler with reasonably correct up and down geometry then the desire to shorten the distance from the front wheels to ‘back of dash’, which would place the engine as far forward as possible, would make it convenient to splay the axes of at least the longer lower levers so as to keep these out of the way of the engine.

The Lagonda layout sprang from a realization that the shorter upper levers were outside the space occupied by the engine. Associated with this was a desire to use a double piston and lever type hydraulic damper actuated by a rod up the middle of the coil spring operated by the lower triangle lever. The upper lever was swung round towards the back far enough to clear this damper operating rod.
An even greater divergence from parallelism was incorporated in the Rover 2000 introduced in 1963 where the upper lever was fore and aft, the transverse axis making it possible to couple an anti-roll rod directly to the levers. In the Nissan multi-link suspension (see Figure 2-22), double wishbone geometry has been ingeniously incorporated within the confined packaging restraints of wide tires and low bodywork by adding an extra curved upper part to the hub carrier. This allows a much longer upper wishbone to be used and greater freedom in the arrangement of the wheel geometry, independent of the king pin axis.

An example of a recent passenger car front suspension is the double wishbone front suspension of the Honda models Prelude and Accord shown in Figure 2-23, with short upper wishbones with widely spaced bearings, lower transverse control arms and longitudinal rods whose front mounts absorb the dynamic rolling stiffness of the radial tyres. The shock absorbers are supported via fork-shaped struts on the transverse control arms and are fixed within the upper link mounts. This point is a good force input node. Despite the fact that the upper wheel carrier joint is located high, which gives favourable wheel kinematics, the suspension is compact and the bonnet can be low to give aerodynamic advantages. The large effective distance between the upper and lower wheel hub carrier joints results in low forces in all mounts and therefore less elastic deflection and better wheel control.

![Honda Prelude and Accord double wishbone suspension](ref.4)
Another view on independent passenger car suspension

Thomas D. Gillespie (ref. [9]) gives another view of the independent suspension on cars;

In contrast to solid axles, independent suspensions allow each wheel to move vertically without affecting the opposite wheel. Nearly all passenger cars and commercial vans use independent front suspensions, because of the advantages in providing room for the engine, and because of the better resistance to steering (wobble and shimmy) vibrations. The independent suspension also has the advantage that it provides inherently higher roll stiffness relative to vertical spring rate. The first independent suspension appeared on front axles in the early part of the twenty century. Maurice Olley (ref. [10]) deserves much of the credit for promoting its virtues, recognizing that it would reduce some of the wobble and shimmy problems characteristic of the solid axles (be decoupling the wheels and interposing the mass of the car between the two wheels). Further advantages included easy control of the roll center by choice of the geometry of the control arms, the ability to control tread change with jounce and rebound, larger suspension deflection, and greater roll stiffness for a given suspension vertical rate.

The most common design for the front suspension of American cars following World War 2 used two lateral control arms to hold the wheels. The upper and lower control arms are usually of unequal length from which the acronym SLA (short-long arm) gets its name. The arms are often called ‘A-arms’ in the US and ‘wishbones’ in Britain. This layout sometimes appears with the upper A-arm replaced by a simple lateral link, or the lower arm replaced by a lateral link and an angled tension strut, but the suspension are functionally similar.

Automobile literature conclusions

The double wishbone suspension geometry on commercial vehicles is almost the same as found on passenger cars. Mean differences are the extra fusees on trucks and the use of anti-features on cars. Variation in types on passenger cars is great, because they use double wishbone suspensions for a long time. Most advantages for passenger cars will also apply to commercial vehicles, the most important ones are:

- Space saving.
- Attenuate of road noise, improving ride comfort.
- Decreasing steering vibration, to prevent wobble and shimmy problems.
- Higher roll stiffness relative to the vertical spring rate.
- Camber change, to minimize track alternation which minimize tyre wear and adapt good camber angles during corning, so the tyres staying horizontal on the road contact patch while the chassis body is rolling over.

The commercial truck industry can learn a lot from the automobile market regarding the double wishbone suspensions, because the automobile market uses this type of suspension for a long time. There is a lot of literature available considering personal vehicles compared to the commercial vehicles. It is recommended using this literature when designing an independent front suspension for a commercial vehicle.
3 IFS design for trucks

In this chapter the independent front suspension design will be explained. First the suspension ride, the suspension handling and the suspension geometry design objectives are summarized. According to these objectives the final design of the independent front suspension for trucks will present. At the end the different design choices such as parallel A-arms, inboard instant centers and no anti-roll bar are explained.

3.1. Design objectives

Before an independent front suspension can design, first the design objectives must clearly state. The design objectives define the most important suspension characteristics. Regarding to those objectives the best design decisions can make. Looking back to the objectives, during the development period prevents loosing the main design points out of sight.

General objectives

Overall:
The IFS is designed as an independent front air suspension module with ride, handling and weight superior to all competitive suspensions. Detailed performance objectives are given below.

Durability:
System durability requirements are set as being equal to or better than the existing ones. Of particular importance is the need to offer a zero maintenance system.

Weight:
Design IFS with low weight comparing to the weight of the original rigid beam suspension.

Simplicity:
A simple design offering ease of assembly and with a reduced number of parts is desirable, together with features that reduce assembly time and costs.

Packaging:
The system must be installed on existing vehicles, taking into account the very large permutation of customer specified components and high powered diesel engines.

Suspension ride objectives

Ride comfort:
The ride comfort of the IFS must be better or at least equal to the original rigid axle configuration. In general, the isolation from road inputs measured at the cab must to be better than the original rigid axle suspension.
Unsprung Mass:

Apart from the general desirability of a low weight design a lower unsprung mass contributes to improving ride comfort. The important components that must be low weight designed are the unsprung elements of the suspension.

Damping:

System damping needs to be optimized to cope with a very low hysteresis suspension while not compromising ride quality.

Wheel travel:

A design requirement comparable to the original rigid axle suspension of the wheel travel is necessary, bearing in mind that its attainment dependent on wheel track and frame width.

Suspension handling objectives

Roll Stability:

The roll stiffness has to be equal to or greater than the original rigid axle suspension that has demonstrated the ability to cope with the high sleeping cabins.

Roll Steer:

Understeer to neutral behavior in roll is mandatory. The exact coefficient must be tuned to the value best suited to the vehicle selected.

Bump Steer, Brake Steer, Wheel Kick and Lateral Stiffness:

For all these characteristics it is decided to use the best well-know characteristics of the original rigid axle suspension and adopt this as a target to equal or surpass.

Suspension geometry

Camber:

Static camber must be the same as the original rigid axle suspension. Camber change is now (in the IFS design) for the first time possible for trucks, the change rate has to be investigated and depends on the suspension arms lengths.

Kingpin:

The kingpin angle and offset must be comparable with the in the truck industry standard values. This to find easy acceptance of the IFS in the truck industry, after acceptance those values can still be optimized.

Caster:

Static caster must also be comparable to the original rigid axle suspension. Although there is caster change possible it is recommendable to minimize it, because caster change will affect the forward driving stability, which is an important issue during driving a truck straight forward.

Steering geometry:

In the truck market is it common to use full Ackermann steering to realize minimum tire wear. Obviously trucks don’t reach high lateral accelerations, but maneuver a lot at low speed.
3.2. IFS design

The independent double wishbone front suspension design exists of an upper and lower A-arm. In the front plane (y-z plane) the A-arms has an inboard instant center (see Figure 3-1), which means there is negative camber change on the outside wheel and positive on the inside wheel during body roll when cornering. This gives the truck better steering properties but also more tire wear, because of track alteration see Figure 3-2.

![Figure 3-1: Technical view with roll center and instant centers (frontview)](image)

The side plane (x-z plane) shows the upper and lower A-arms are installed parallel to each other (see Figure 3-3), so there is no anti-dive effect during braking. Because of the high center of gravity of a truck there will be a high level of load transfer to the front axle when applying the brakes, so an anti-dive geometry could be beneficial. Advantage is the simple steering design when using parallel A-arms.

![Figure 3-2: Camber change during suspension bump & roll motion](image)

The uprights do not allow rotation around the vertical z-axis because of the used joints so additional fusees are necessary to turn the wheels. It is very difficult to install rotational uprights, with the king-pin axle placed near towards the wheel center, because of the relative small inside radius of the 22.5 inch wheel rims. Also the upright has to be very small to fit into the rims. Small uprights which have a short vertical distance between the upper and lower A-arms connections would be responsible for big lateral forces into the A-arms. Also when the king-pin axle is located further away form the wheel center to the inside, the king-pin angle and scrub...
radius will increase, this has bad consequences to the steering properties (see section 2.1 Front suspension kinematics). Expensive and less reliable spherical ball joints are necessary to connect the A-arms to the rotational uprights to allow steering the uprights. Using fusees and parallel mounted A-arms makes it possible to use simple robust revolution joints which only are able rotate around one axis, instead of three. It may be hard to install this system because the A-arm must mount exactly parallel, because the system is essentially over-constrained. Using an extra turning axis (fusee) will increase the unsprung mass, which has a bad influence on the ride comfort.

![Figure 3-3: Parallel longitudinal A-arms, so no anti-features possible (sideview)](image)

The springs and dampers are connected to the uprights (see Figure 3-4); this is possible because there is no rotate of the uprights during wheel turning, this is also a reason to use separate fusees. There are no (negligibly low) spring and damper rates when mounting the springs and dampers to the uprights. The springs and dampers will exactly following the vertical suspension displacement. When mounting the spring and dampers to the upper or the lower A-arms the spring and damper rate will increase when mounted closer to the chassis. Only non-linearity which will occur is the change in the angle of the spring and damper during vertical wheel travel due the rotational movements of the A-arms. Placing the springs and dampers as vertical as possible will help minimizing the non-linearity during wheel travel.

![Figure 3-4: Spring and damper displacement during wheel travel](image)
An anti-roll bar is not necessary, because the springs are effectively located at the same track width as the wheels. This large spring track width results in high roll stiffness while the bump stiffness remains the same because this is dependent on both springs. The original rigid axle leafspring suspension has a shorter track width with less roll stiffness, because the springs are mounted under the longitudinal runners of the truck chassis. Saving an anti-roll bar will reduce the unsprung mass and number of parts. A disadvantage which will appear in chapter 5 is single bump will introduce higher forces into the truck chassis during roll motions.

The used steering system on the IFS is a so-called rack and pinion steering system which is often used on passenger cars. This system operates using a translational movement; the rack will move to the right or left when the pinion is rotating. The pinion is connected to the steering wheel and the rack to the steering rods (tie-rod), which turn the wheels for steering. During the design of the steering system the tie-rods length and location are very important (see section 2.1). Figure 3-5 shows the geometric construction of the length of the tie-rod for given A-arms lengths. The IFS A-arms are parallel in longitudinal (x-) direction; this means only looking to the front plane (y-z plane) is enough for designing the right tie-rod configuration. The aim is to have no undesirable bump steer.

The IFS system is designed to create more engine installation space, so a bigger engine can be installed. The IFS is, with all its components such as the sub-chassis, heavier in comparison to the original rigid axle suspension. The unsprung mass of the IFS is, because using extra fusees, not much lower comparing to the rigid beam. This restricts the possible ride comfort improvements due to weight reduction.
IFS design for trucks
4 MATLAB multi-body model

The existing tractor semi-trailer multi-body model (ref. [11]) is used as a basis for the tractor model with an independent front suspension. The truck library is extended with an IFS block, so the IFS and original rigid front axle suspension can easily be exchanged. In this chapter an overview will give of how the model is build up.

4.1 Tractor semi-trailer

The original tractor is divided in five main parts; the cabin, the driveline, the front axle, the rear axle and the two-parts tractor chassis. The semi-trailer exists of three trailer axles, a three-part trailer chassis and three loads as shown in Figure 4-1.

Figure 4-1: Tractor semi-trailer structure overview
Tractor semi-trailer vehicle model parameters

Sign convention:

- **X**: positive in the driving direction, \( X=0 \) centre of the front wheels
- **Y**: positive to the left, \( Y=0 \) plane of symmetry of the vehicle
- **Z**: positive upwards, \( Z=0 \) road level

Units:

In the model SI units (m, kg, N, s, rad) are always used.

General:

- The semi-trailer is a generic three axle configuration.
- Coordinates are given for the vehicle in a fully loaded condition.

Main dimensions:

- **Front axle**: \( X = 0 \) m
- **Rear axle**: \( X = -3.8 \) m (wheelbase: \( WB = 3.8 \) m)
- **5th wheel**: \( X = -3.13 \) m, \( Z = 1.15 \) m (kingpin - rear axle: \( KA = 0.67 \) m)
- **1st trailer axle**: \( X = -9.63 \) m (distance to 5th wheel = \( 6.5 \) m)
- **2nd trailer axle**: \( X = -11.03 \) m (distance to 5th wheel = \( 7.9 \) m)
- **3rd trailer axle**: \( X = -12.43 \) m (distance to 5th wheel = \( 9.3 \) m)
- Overall length of the vehicle combination: \( 15.50 \) m

Tractor semi-trailer

Engine/driveline:

Engine and gearbox represented in Figure 4-2 are modelled as a single rigid body, elastically suspended in all 6 degrees of freedom with respect to the chassis.

The drive torque (see Figure 4-3) is calculated based on engine power and angular velocity and scaled using the throttle input. The driveshaft is massless and infinitely stiff.

The drive torque is applied to a revolute joint between engine and driveshaft and (taking into account the final drive ratio and differential) to the wheels.

Cabin:

The cabin is modelled as a single rigid body, elastically suspended in all 6 degrees of freedom with respect to the chassis (see Figure 4-5). The anti-roll bar is modelled as a torsional spring in the X direction. Aerodynamic forces on the cabin are not considered.
Front axle:

The front axle is modelled as a single rigid body having 2 degrees of freedom with respect to the chassis; bounce and roll (see Figure 4-4). The anti-roll bar is modelled as a torsional spring in the roll degree of freedom. Roll steer and steering system kinematics (caster, KPI, etc.) are not considered.

Rear axle:

The rear axle is modelled in the same way as the front axle.

Tractor chassis:

The chassis is modelled as two rigid bodies (see Figure 4-6), connected by a revolute joint to account for flexibility (torsion).

Trailer axle:

The trailer axle is modelled in the same way as the front axle

Trailer chassis/Load:

The chassis (see Figure 4-7) is modelled as three rigid bodies, interconnected by a revolute joint to account for flexibility (torsion). The 5th wheel is connected to chassis body 1, axle 1 is attached to chassis body 2 and axles 2 and 3 are attached to chassis body 3. The loads 1, 2 and 3 are rigidly connected to the respective chassis bodies.
4.2. **IFS SimMechanics library block**

The existing tractor semi-trailer truck library is extended with a new library block; the Independent Front Suspension "IFS". The IFS multi-body model exists of other blocks, bodies and joints. The brake wheel bearing and tires are imported from the existing truck library. A tractor semi-trailer model with independent front suspension can be built by changing the normally used front axle (rigid axle leafspring suspension) with the IFS. In this paragraph the IFS model block will be explained.

**IFS model structure**

A short overview of the model structure is shown in Figure 4-8. The different parts used to construct the IFS model will be explained in this paragraph (see Figure 4-9 for a technical view of the IFS model.

![Figure 4-8: IFS SimMechanics model](image-url)
8.4 Ride comfort weighting functions

The A-arms

The IFS is a double wishbone suspension, which means the model is built up of four wishbones also called A-arms (see Figure 4-8). Lower A-arms (see Figure 4-10) are mounted to a sub frame, which is specially designed for the lower A-arms. The upper A-arms (see Figure 4-11) are mounted to the longitudinal chassis members. The upper and lower A-arm on the left are mirrored to the right. The A-arms are connected to the chassis with revolution joints, which are responsible for a restricted motion of the A-arms in only the y-z plane. The rotation axes of the upper and lower A-arm revolution joints mounted to the chassis are parallel to each other. To prevent an over constrained model, one of the corners of the parallelogram must have an additional degree of freedom. Therefore the upper A-arm is equipped with a spherical joint which can rotate freely about all three axes and has an additional translational degree of freedom in x-direction.

Uprights and fusees

The A-arms are connected to an upright (see Figure 4-8). The upright can not rotate around the vertical z-axle because connection to the lower A-arms with revolute joints. Those revolute joints only have a degree of freedom around the x-axle. The A-arms are located parallel to each other (no anti-effects possible) and the translation is only vertical (no caster change possible). It is not possible to use features like caster change and anti-dive. The uprights have an internal
revolution joint (the fusees) to create wheel turning on the frontaxle. These internal revolution joint can rotate around the z-axle with a static KPI, caster and camber angle. Normally the fusees are connected to the A-arms by spherical bal-joints. Because the high level of forces on the joints it is difficult to use bal-joints on a truck suspension, eliminating internal fusees makes the suspension less heavy. On the other end of the fusees the wheel hubs are connected.

**Anti-roll bar**

There is an anti-roll bar (see Figure 4-8) modeled, but the torsion stiffness of the anti-roll bar is set to zero. This can occur because the air springs are effectively located on the IFS virtual frontaxles' track width, which create enough roll stiffness. The torsion stabilizer is modeled as a virtual rigid beam; on its center of gravity point the roll and bump movements will be measured and tells something about the difference in acceleration and motion of the IFS comparing to the rigid frontaxle suspension. The stabilizer exist of a body that only can translate in vertical direction (bump) and rotate around the x-axle (roll), this body is connected to the uprights with two rods that can change in length using a translation joint to deal with the track alternation.

**Wheels**

The wheel hubs are connected to the brake/wheel bearings (see Figure 4-8) that are connected to the tires, both brake/wheel bearings and tires blocks are the same used in the original truck model with the rigid frontaxle suspension. The brake/wheel bearings are actuated by brake moments and connected to the tyres.

**Springs and dampers**

Between the dummy chassis and the uprights the non-linear airsprings and dampers (see Figure 4-8) are mounted. Because they are separated parts placed on separated places the original spring-damper block is split in two parts, a spring and a damper block.

**Steering**

With use of a translation joint the steering rack is modeled in the IFS model (see Figure 4-8). The body which moves in y-direction turns the wheel hubs by a spherical – spherical joint. The translation joint is actuated by the steering input.

**Chassis**

The complete IFS model block is connected to the tractor chassis of the original truck model. By using different blocks the exchange of the IFS with the original rigid frontaxle suspension is very easy; furthermore the same model blocks are used to create the remaining part of the truck.
IFS block settings

To analyze IFS with different parameters settings scaling factors have been built into the model. Parameters that can be set are the spring stiffness, damping, unsprung masses and chassis torsion stiffness.

The airspring stiffness and damping

The spring stiffness is dependent of the air pressure in the airsprings. The air pressure stiffness can be set with a spring factor. This also can be done for the damping with the damping factor. The spring and damping factors are part of the block parameter box of the IFS.

The unsprung mass

Also the unsprung mass can be changed. With a mass factor the unsprung mass of the IFS can be decreased or increased.

The chassis torsion stiffness

To see what the influence of the chassis torsion stiffness is, the torsion stiffness can easily be changed by a multiplication factor in the block parameters box.
MATLAB multi-body model
5 Ride comfort comparison

Before looking at the simulation results first the quantity “ride comfort” and the calculation of the Ride Comfort Index abbreviated as “RCI” are explained and the simulation conditions and parameters of the tractor semi-trailer model will be discussed. The simulation results of the original rigid front axle suspension are compared to the new IFS. The influence of chassis torsion stiffness is analyzed in the same way as the sensitivity of spring stiffness, damping and unsprung masses. This chapter will give an overview of the necessary adaptations like a stiffer chassis and modified spring and damper characteristics, when equipping a tractor with IFS.

5.1. Comfort

The following definition of comfort can be given;

Comfort: 1: a state of being relaxed and feeling no pain,
2: a feeling of freedom from worry or disappointment,
3: the act of consoling; giving relief in affliction

Comfort has both psychological and physiological components, but it involves a sense of subjective well-being and the absence of discomfort, stress or pain (Richards 1980, ref. [13]). However, comfort is not only defined by the absence of negative attributes. It is possible to feel a positive experience of comfort to various degrees. Comfort involves evaluation; it is felt to be good and its opposite is felt to be bad. The only way to of finding whether a person is comfortable or not is ask the person in question. Comfort in transportation research is normally defined as subjective well-being, although comfort is one of the variables that may contribute to well-being, it is not a necessary part of it. However, there exists many terms, such as comfort, passenger comfort, ride quality, ride comfort, ride index, that are not clearly defined and which have been used with varied and overlapping meanings. The sections below are an attempt to structure the terminology concerning comfort.

The ride quality

Ride quality is a person’s reaction to a set of physical conditions in a vehicle environment, such as dynamic, ambient and spatial variables. Dynamic variables consist of motions, measured as accelerations and changes (jerk) in accelerations in all three axes (lateral, longitudinal and vertical), angular motions about these axes (roll, pitch and yaw) and sudden motions, such as shocks and jolts. Normally, the axes are fixed to the vehicle body. The ambient variables may include temperature, pressure, air quality and ventilation, as well as noise and high frequency vibrations, while the spatial variables may include workspace, leg room and other seating variables. Other factors may be convenience of the transport, frequency, etc (Richards 1980, ...
ref. [13]). However, many use the term passenger comfort, ride comfort or average ride comfort for ratings on a ride quality scale regarding the influence of dynamic variables. Normally, higher rating on a ride quality scale means better comfort, whereas higher rating on a ride (dis-)comfort scale means less comfort.

**The ride comfort**

In this study, ride comfort, or more precisely ride discomfort, will be used as the technical evaluation of dynamic quantities (motions of the vehicle). This is in accordance with (ISO 2631-1, 1997; ref. [14]). This technical evaluation is based upon human reactions to these dynamic quantities. It is about isolating the driver from vibrations occurring as a result of operating the vehicle (sources: road, tires, engine, driveline etc.). When driving, the vehicle experiences a broad spectrum of vibrations. These vibrations are transmitted to the passengers either by tactile, visual or aural paths. In general the term 'ride' is used in reference to tactile and visible vibrations, while the aural vibrations are characterized as 'noise'. In the frequency domain, we may classify the spectrum of vibrations below 80 Hz important for ride and that from 25 Hz to 20 kHz as noise. The 80 Hz boundary is the upper limit of the frequency range according to which the vibration comfort of the human body is usually assessed (ISO 2631-4, 2001; ref [15]). The 25 Hz and 20 kHz boundaries are approximately the lower and upper frequency thresholds of human hearing.

**Influence from vehicle, track, other physical factors**

**And human factors on ride comfort and ride quality**

![Figure 5-1](Image)

Figure 5-1: Interaction in the field of ride comfort and ride quality evaluation (ref. [17])
Other denominations used for ride comfort are passenger comfort (Oborne D.J., ref. [16]) and ride quality (Richards 1980, ref. [13]). The human evaluation of comfort, ride quality, involves not only motion quantities but also human interaction variables, see Figure 5-1. This agrees with the evaluation of ride comfort in Japan. Human variables may be of a social or situational kind. Modifying human variables include age, gender, posture, alcohol, experience and mental activity [Griffin 1990, ref. [17]]. Psychological variables are important and can modify the severity of the human responses. Among these variables are expectation and suggestion, specific conditioning effects of past experiences, habituation effects of past experience, effects of concurrent activity and effects of concurrent emotional state.

**ISO weighting functions**

Due to its subjective nature ride comfort is difficult to asses. Nevertheless, many recent research results show a minimum tolerance (maximum sensitivity) of the human body to vertical vibrations in the frequency range between 4 and 8 Hz. This sensitivity is the result of vertical resonances of the abdominal cavity. This is shown in Figure 5-2, where the human body is depicted as a multibody system. At frequencies above and below this range, the tolerance increases in proportion to frequency.

![Figure 5-2: Human body sensitivity for specific frequencies (ref. [18])](image)

The overall ride comfort including the human vibration sensitivity will be calculate according to ISO 2631-1, 1997 weighting functions described in appendix D.
5.2. Simulation conditions

The simulation conditions as described below are the same for the rigid front axle suspension and the IFS configuration.

To calculate the Ride Comfort Index the tractor semi-trailer is simulated under the following conditions:

- Total weight of the truck-trailer combination is ±40000 kg, which corresponds to a fully loaded vehicle.
- A measured road profile, classified as good following [ISO 8608:1995(E)], is used as road during the simulations (see appendix B).
- The TNO (semi-) empirical tire model is used for simulating the tires, which is an implementation of the Magic Formula (see appendix C).
- Simulation time is 60 seconds, in the first 5 seconds the tractor semi-trailer is settling for static equilibrium. The other 55 seconds it’s driving over the road profile.
- The tractor semi-trailer is traveling with a forward velocity of 85 km/h and is controlled by a cruise control.
- No steering input is given which means the tractor is driving in a straight line, so there are no handling maneuvers during the ride comfort simulations.
- No camber, caster and KPI angle are applied, because there is no steer input it is not necessary to insert these parameters, so in this simplified simulation they are all set to zero.
- Only the front axle block will be exchanged, the single simulation model difference between the IFS and the rigid front axle suspension is the used front axle multi body block.

These are the simulation conditions used during the ride comfort investigation, unless described otherwise these conditions also are used in all other simulation during this master thesis.

5.3. Comparing IFS to rigid front axle suspension

The most important simulation result is the acceleration of the driver. This acceleration gives the magnitude of vibration when driving over a given road profile. Ultimately the driver is sitting on the drivers’ seat and experiences the vibrations and so the driver will classify the ride comfort. Therefore the accelerations of the in the cabin placed drivers seat will be measured. Weighting the accelerations according to the ISO weighting function will lead to the ride comfort index called RCI. First the RCI of IFS will be compared to the RCI of the rigid front axle suspension, which shows the difference in ride comfort.
8.4 Ride comfort weighting functions

Simulation results

<table>
<thead>
<tr>
<th>Parameters:</th>
<th>sensor locations:</th>
<th>Leafspring</th>
<th>Leafspring with IFS parameters</th>
<th>IFS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ride Comfort Index (RCI)</td>
<td>driver’s seat</td>
<td>0.51 m/s²</td>
<td>0.51 m/s²</td>
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<td>Longitudinal x-acc. (RMS)</td>
<td>cabin’s CG</td>
<td>0.35 m/s²</td>
<td>0.31 m/s²</td>
<td>0.30 m/s²</td>
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<td>Lateral y-acc. (RMS)</td>
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<td>0.50 m/s²</td>
<td>0.92 m/s²</td>
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<tr>
<td>Vertical z-acc. (RMS)</td>
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<td>34143 N</td>
<td>33941 N</td>
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<tr>
<td>Suspension travel (RMS)</td>
<td>frontaxle</td>
<td>4.4 mm</td>
<td>6.4 mm</td>
<td>8.6 mm</td>
</tr>
</tbody>
</table>

Table 5-1: RCI Simulation results

Table 5-1 presents the RCI and the most important RMS values of the rigid frontaxle suspension and IFS. The RCI of the rigid frontaxle suspension is quite low and can be quantified as “a little uncomfortable” according to [ISO 2631-1, 1997] shown in appendix B. The IFS has a higher RCI value and can be quantified as “fairly uncomfortable”. The negative result of the IFS ride comfort is an unexpected result, after all it was expected that an independent suspension would increase the ride comfort. The RMS values of the Cabin CG shows that the biggest difference occurs in the lateral acceleration, which is almost twice as high for the IFS.

To understand the difference in the ride comfort index between the IFS and rigid frontaxle suspension some PSD's are plotted. Those power spectral densities (PSD) show the strength of the variations (energy of vibrations) as a function of the frequency. In other words, it shows at which frequencies variations are strong and at which frequencies variations are weak.

In the PSD figures three simulation configurations are plotted:

1. The original rigid frontaxle suspension tractor semi-trailer with realistic spring, damper and anti-roll parameters validated with measurement results of a real tractor semi-trailer truck combination (bleu line).
2. The original rigid frontaxle suspension equipped with spring and damper parameters of the IFS suspension. Because the big differences between the spring and damper parameters of the rigid frontaxle suspension and the IFS, this additional simulation configuration is added for a more useful comparison (green line).
3. The independent front suspension (IFS) tractor semi-trailer with the concept design spring and damper parameters and a zero torsion anti-roll bar (red line).
Ride comfort comparison

Figure 5-3: x, y and z-accelerations PSD plot of the cabin CG and driver’s seat
Figure 5-4: x, y and z-accelerations PSD plot of the driver’s seat and the weighted driver’s seat.
Low frequencies (< 5 Hz)

Looking at the x, y, and z-acceleration PSD’s of the cabin CG point, the drivers’ seat and the weighted drivers’ seat shows some interesting results (see Figure 5-3 and Figure 5-4). The z-acceleration PSD shows some peaks in the low frequency range 0.1 Hz to 4 Hz, which are responsible for the low cabin pitch mode around 1.8 Hz. The differences between the IFS and the rigid frontaxle suspension (configurations 1 - 3) in this low frequency range are quite small and mainly caused by the difference in spring and damper characteristics. Furthermore, these differences will be filtered out due to the weighted function (explained in appendix D). Looking at the weighted driver’s seat, which shows the driver’s seat movements corrected with the ride comfort weighting functions (see Appendix 8.4), the x and z-accelerations in the low frequency range are negligibly low. It can be concluded the problem is not caused by the low frequencies.

High frequencies (> 5 Hz)

Looking at the y-acceleration and z-acceleration of the cabin CG the IFS configurations (conf. 3) shows some large peaks in the high frequency range from 7 Hz to 10 Hz, those peaks are less high on the rigid frontaxle suspension (conf. 1 and 2). This means there is a lot of roll in the IFS model. Looking to the weighted seat y-acceleration this motion is reduced in amplitude to low acceleration values, which is caused by the decrease of the longitudinal & lateral weighting function plot by higher frequencies. The z-acceleration of the driver’s seat of the IFS shows, at high frequencies, the same high accelerations as in the cabin CG y-acceleration plot. The cabin roll is responsible for motion of driver’s seat in both y and z-directions; this means the roll motion of the cabin CG will create a lot of vertical accelerations to the driver’s seat. The vertical weighting function will increase by higher frequencies. For this reason the weighted IFS driver’s seat z-acceleration PSD plot shows high acceleration, unlike the rigid frontaxle suspension. This explains the high frequency roll is the main reason of the difference in ride comfort between the IFS and the leafspring suspension.

High frequency roll

What causes the high frequency roll motions incase of the IFS? To get an answer on this question other parts of the truck have to be investigated, especially the ones which are in contact with the IFS itself. Therefore the roll and bump motion of the front chassis, driveline and frontaxle are plotted in Figure 5-5. Looking at the rotation acceleration (Φddot) it is clear that the high frequency roll around the 10 Hz is started on the front axle and transfer due to the driveline and front chassis to the cabin. The rigid frontaxle suspension roll is less high and spreads over a larger frequency range, unlike the IFS which has large amplitudes over a small frequency range. The rigid frontaxle suspension roll is damped out more over the differed parts, while the large roll amplitudes of the IFS are losing less activity. The difference in roll motion is visible on the front chassis in an increase state comparing roll of the frontaxle itself. This means the IFS suspension introduces more roll to the front chassis, which is coupled directly to the cabin, comparing to the rigid frontaxle suspension. So the bad ride comfort of the IFS would be caused by the difference in the front suspension construction.
Figure 5-5: bump and roll motion PSD’s of the front chassis, driveline and frontaxle
Looking at the road profile (see Appendix B) it clearly shows the long wave height differences, caused by road sinks, are responsible for symmetric two wheel bump (both left and right wheel will be actuated at the same time). This means that in the low frequency range both wheels travel vertically. The high frequency wheel motions will be created by put holes, short and small road damage, which are uneven and responsible for one wheel bumps (only left or right wheel are actuated). This means the high frequency roll motion problem is caused by asymmetric wheel actuation.

Looking at the design of the IFS and rigid front axle suspension there are two important differences, namely the difference in track width and resulting mutual wheel influence. By symmetric road excitations the front wheels will travel parallel, so both left and right spring and damper will be actuated at the same time and level (see Figure 5-6 above). In the other situation by asymmetric road excitation the IFS and rigid front axle suspension show differences. When one wheel bump occurs on the IFS only the spring and damper on the traveling wheel (left or right) will be actuated. The level of spring and damper actuation is linear with respect to the level of the suspension travel. In the IFS design the spring and damper are connected to the upright which follows the path of the wheels exactly. On the rigid front axle suspension the level of spring damper actuations differ a lot. When one wheel bump occurs to the rigid front axle suspension both left and right springs and dampers will be actuated due to the rigid axle beam (see Figure 5-6 below). The level of actuation of the left and right springs and damper are not the same and depend on the track width of the wheels, springs and dampers. On the IFS the track width of the springs and dampers are theoretical the same as the track width of the wheels, but on the rigid front axle suspension those are smaller. The IFS chassis has to deal with higher spring and damper forces because an asymmetric one wheel bump will act to only one (left or right) longitudinal chassis beam. On the rigid front axle suspension one wheel bump will act on both longitudinal chassis beams, so the chassis roll motion caused by asymmetric road excitations will be higher on the IFS compared to the rigid front axle suspension.
Conclusion

The bigger roll motion which is responsible for the bad ride comfort of the IFS is caused by the difference in one wheel suspension travel. During one wheel bump both left and right springs and dampers are actuated on the rigid front axle suspension, while only the left or right wheel spring and damper will be actuated on the IFS. This creates a difference in forces acting on the chassis. The IFS brings more roll motion into the chassis which will give also more roll motion in the cabin and driver’s seat where finally a decrease in ride comfort will be measured.

5.4. Chassis torsion stiffness influence

Because of the importance of the roll motion and the suspension design to the ride comfort, it is interesting to look at the chassis torsion stiffness influence. The tractor chassis takes care of the connection of the front axle to the other parts of the truck like the engine, cabin and rear axle. In the multi-body tractor model the chassis is divided into two parts; the front and rear chassis. The two chassis parts are connected by a revolute joint to create flexibility (torsion). By changing the value of the torsion stiffness the influence of the torsion to the ride comfort can be analyzed. In Figure 5-7 the torsion stiffness versus the ride comfort is plotted.

Figure 5-7: Chassis torsion stiffness influence to the ride comfort index RCI
Figure 5-7 shows on the x-axis the chassis torsion stiffness and on the y-axis the ride comfort index RCI. The values on the x-axis are the multiplication factors of the original chassis torsion stiffness. The plot shows a high decrease of the RCI (lower RCI values = better ride comfort) of the IFS configuration by increasing the chassis torsion stiffness. The RCI decrease of the rigid front axle suspension configuration is less high. In comparison to the rigid front axle suspension, the IFS show a high influence of chassis torsion stiffness. Figure 5-7 shows that the IFS will intersect the rigid front axle suspension and become the one with the best ride comfort, but this occurs at 40 times the original normal chassis torsion stiffness. Such an increase of chassis torsion stiffness is impossible to realize with chassis used nowadays. Higher increases as 3 times the original chassis torsion stiffness are technically very difficult to design. The torsion stiffness influence plots shows the increasing of the chassis torsion stiffness to 3 times the original value will not significantly improve the ride comfort of the IFS. The dynamic wheel load will increase slightly when the RCI decreases, this is caused by the decrease of roll motion of the front chassis when the torsion stiffness increased. Also the spring and dampers will have less freedom to travel, when the chassis roll decreases.

Figure 5-8: Roll PSD’s by 1x and 25x the chassis torsion stiffness
In Figure 5-8 the PSD’s the torsion stiffnesses of 1 time and 25 times the normal torsion stiffness are plotted for the cabin, front chassis, driveline and front axle centers of gravity. It shows clearly that the high peaks around the 9-10 Hz of the IFS disappear. Decrease of the high frequencies is the main reason of improvement of the RCI using a stiffer truck chassis. For the rigid front axle suspension also a slight decrease of the PSD is shown around 9-10 Hz. In the plot of the cabin roll a peak shows up around 3.2 Hz by the 25 times higher torsion stiffness. Interesting is the similarity of the 25 times torsion stiffness IFS plot with the 1 times torsion stiffness rigid front axle suspension plot. Conclusion is the regular rigid front axle chassis roll motion is comparable to the chassis roll motion of the IFS with a 25 times stiffer chassis.

Conclusion

The IFS ride comfort is highly dependent on the chassis torsion stiffness. Nowadays a truck has a very flexible chassis existing of two mean longitudinal runners and some crossbeams. The possibilities to increase the chassis torsion stiffness are very small. This means the IFS ride comfort improvement with regard to chassis torsion stiffness only can realize using a completely different chassis type. A good example is a touring bus which uses independent suspension in combination with a stiff space frame chassis (integral-frame construction).

5.5. Sensitivity investigation

A sensitivity investigation is carried out to investigate the influence of the other three most important parameters on the ride comfort; the spring stiffness, the damping and the unsprung masses. The spring stiffness and damping are linearized to prevent non-linear behavior. The IFS is equipped with the rigid front axle suspension’s linear spring and damper characteristics, so differences in spring and damping parameters will not influence sensitivity investigation. This makes it possible to make a fair comparison between the rigid front axle suspension and the IFS. The rigid front axle suspension stiffness and damping are averaged between the original IFS and the original rigid front axle suspension values. The spring stiffness, damping and unsprung masses are simulated from 30% to 200% of the normal 100% value.
Spring stiffness sensitivity

The plots below show what is happening to the Ride Comfort Index (RCI) when the spring stiffness is varied between 30% and 200%. Increasing the spring stiffness will lower the ride comfort for both the IFS and rigid front axle suspension.

Figure 5-9: Spring stiffness sensitivity plots

Although the IFS shows a very slight RCI decrease (lower RCI values mean higher ride comfort) between 30% and 70%, its RCI increases significantly between the normal 100% and 200%. The increase of the rigid front axle suspension RCI is even more in comparison to the IFS, so changing the spring stiffness will have more effect on the rigid front axle suspension. The dynamic wheel load of the IFS will increase more than on the rigid front axle suspension. The vertical wheel displacement logically decreases when raising the spring stiffness.
**Damper sensitivity**

Also looking at the damping, varied between 30% (softer) and 200% (harder), an increase in damping will lead to an increase of the RCI for both the IFS and rigid front axle suspension (see Figure 5-10). This means softening the dampers will improve the ride comfort for both the IFS and rigid front axle suspension.

If the damping decreases the RCI value of the IFS remains decreasing, the rigid front axle suspension will stabilize below 60%. Consequence is converging of the IFS to the rigid front axle suspension as shown in the RMS cabin CG plot. The dynamic wheel loads will increase a bit when the ride comfort becomes better. The wheel travel increases significantly when softening the dampers. The IFS and rigid front axle suspension can better absorb road disturbances when softening the dampers to a certain value, this benefits the ride comfort of both the IFS and rigid front axle suspension.
Sensitivity of the unsprung masses

Changing the unsprung masses have high influence on the ride comfort of the IFS, unlike the rigid frontaxle suspension where the effect is very small (see Figure 5-11).

Lowering the unsprung masses will improve the ride comfort of the IFS a lot while the improvement of the rigid frontaxle suspension is relatively low. The ride comfort of the IFS will become comparable to the rigid frontaxle suspension. The suspension is better capable of following the road disturbance when the traveling masses of the IFS are less high. The lower unsprung masses take care of reduced forces to the truck chassis. Both for the IFS and rigid frontaxle suspension the relative wheel travel, the wheel travel with respect to the chassis, will decrease when lowering the unsprung masses, which results in a decrease of chassis roll.
Conclusion

- The spring stiffness has a low influence on the ride comfort for both the IFS and rigid frontaxle suspension, changing the stiffness will not lead to significant improvement of the RCI.
- Changing the damper characteristics can improve both the IFS and rigid frontaxle suspension ride comfort.
- Lowering the unsprung mass improves the RCI of the IFS, unlike the rigid frontaxle suspension where the unsprung mass has a lower influence.

Taken this conclusion into account changing the damper characteristic is the best option for improving the ride comfort of the IFS. Although the improvements are not as high as when lowering the unsprung masses, the ride comfort index changes are significant. Also today there are all kind of technical innovations which can completely changes the behavior of the dampers.
6  Ride comfort improvements

In chapter 5.5 the ride comfort sensitivity of the suspension spring stiffness, damping and unsprung masses are investigated. The sensitivity with respect to the spring stiffness is negligibly low compared to the sensitivity of the damping and unsprung masses. Because reduction of unsprung masses is technically far more complicated compared to changing the damping characteristics, the remaining improvable quantity is the shock absorber.

6.1. Different damper designs

Basic twin-tube design
The twin tube design has an inner tube known as the working or pressure tube and an outer tube known as the reserve tube. The outer tube is used to store excess hydraulic fluid. Rubber bushings are used between the shock absorber and the frame or suspension to reduce transmitted road noise and suspension vibration. The rubber bushings are flexible to allow movement during suspension travel. Twin-tube damper work effectively where there is limited working space. The tubes are designed with smaller diameter. The use of two valves increases the effectiveness of the damper. The valve fitted with the piston rod deals with the rebound control and the inner tube valve deals with bump control. The oil is forced in the outer reservoir in bump mode whereas in the rebound mode, it is pushed through the piston-mounted valve.

Figure 6-1: Twin-tube damper (ref. [15])

A short explanation of the damper working principle:
The piston rod passes through a rod guide and a seal at the upper end of the pressure tube. The rod guide keeps the rod in line with the pressure tube and allows the piston to move freely inside. The seal keeps the hydraulic oil inside and prevents contamination from the outside. The base valve located at the bottom of the pressure tube is called a compression valve. It controls fluid movement during the compression cycle. The bore size is the diameter of the piston and the inside of the pressure tube. Generally, the larger the unit, the higher the potential control levels because of the larger piston displacement and pressure areas. The larger the piston area, the lower the internal operating pressure and temperatures. This provides higher damping capabilities. Engineers select the mounted values for a particular vehicle to achieve optimal ride characteristics, a balance between road holding (stability) and ride comfort, under a wide variety of driving conditions. The selection of valve springs and orifices control fluid flow within the unit determines the feel and handling of the vehicle.
**Twin-tube low pressurized gas charged design**

The design of twin tube gas charged shock absorbers solves many of today's ride control problems by adding a low pressure charge of nitrogen gas in the reserve tube. The gas serves several important functions to improve the ride control characteristics of a damper. The prime function of gas charging is to minimize aeration of the hydraulic fluid. The pressure of the nitrogen gas compresses air bubbles in the hydraulic fluid. This prevents the oil and air from mixing and creating foam. Foam affects performance because it can be compressed - fluid can not. With aeration reduced, the shock is able to react faster and more predictably, allowing for quicker response time.

An additional benefit of gas charging is that it creates a mild boost in spring rate to the vehicle, which reduces body roll, sway, brake dive, and acceleration squat. This mild boost in spring rate is also caused by the difference in the surface area above and below the piston. With greater surface area below the piston than above, more pressurized fluid is in contact with this surface. This is why a gas charged shock absorber will extend on its own.

The final important function of the gas charge is to allow engineers greater flexibility in valving design. In the past such factors as damping and aeration forced compromises in design.

**Advantages:**
- Improves handling by reducing roll, sway and dive
- Reduces aeration offering a greater range of control over a wider variety of road conditions as compared to non-gas units
- Reduced fade - shocks can lose damping capability as they heat up during use. Gas charged shocks could cut this loss of performance, called fade

**Disadvantages:**
- Can only be mounted in one direction upside down

**Twin-tube PSD Design**

With the advent of gas charging, ride engineers were able to open up the orifice controls of these valves and improve the balance between comfort and control capabilities available in traditional velocity sensitive dampers. A leap beyond fluid velocity control is an advanced technology that takes into account the position of the valve within the pressure tube. This is called Position Sensitive Damping (PSD). The key to this innovation is precision tapered grooves in the pressure tube. Every application is individually tuned, tailoring the length, depth, and taper of these grooves to ensure optimal ride comfort and added control. This in essence creates two zones within the pressure tube.

![Figure 6-2: Twin-tube PSD (ref. [15])](image-url)
The first zone, the comfort zone, is where normal driving takes place. In this zone the piston travel remains within the limits of the pressure tube's mid range. The tapered grooves allow hydraulic fluid to pass freely around and through the piston during its midrange travel. This action reduces resistance on the piston, assuring a smooth, comfortable ride which improves the overall ride comfort. The second zone, the control zone, is utilized during demanding driving situations. In this zone the piston travels out of the mid range area of the pressure tube and beyond the grooves. The entire fluid flow is directed through the piston valves for more control of the vehicle's suspension. The result is improved vehicle handling and better control without sacrificing ride comfort.

**Advantages:**
- Allows ride engineers to move beyond simple velocity sensitive valves and use the position of the piston to fine tune the ride characteristic.
- Adjusts more rapidly to changing road and weight conditions than standard shock absorbers
- A shock absorber with both comfort and control characteristics into one

**Disadvantages:**
- If vehicle ride height is not within manufacturer's specified range, piston travel may be limited to the control zone

**Twin-tube ASD Design**

Another alternative shock absorber which provides greater control for handling while improving ride comfort is called Acceleration Sensitive Damping (ASD). This technology moves beyond traditional velocity sensitive damping to focus and address impact. This focus on impact is achieved by utilizing a new compression valve design. This compression valve is a mechanical closed loop system, which opens a bypass to fluid flow around the compression valve. This new application specific design allows minute changes inside the pressure tube based on inputs received from the road. The compression valve will sense a bump in the road and automatically adjust the shock to absorb the impact, leaving the damper with greater control when it is needed. Due to the nearly instantaneous adjustment to changes in the road's condition, the vehicle weight transfer is better managed during braking and turning. This technology improves driver control by reducing pitch during braking and roll during turns.

**Advantages:**
- Control is enhanced without sacrificing driver comfort
- The valve automatically adjusts to changes in the road condition
- Reduces ride harshness

**Disadvantages:**
- Limited availability

Figure 6-3: Twin-tube ASD piston (ref. [15])
Ride comfort improvements

**Twin-tube FSD Design**

Like every twin-tube damper the piston moving back and forth through the oil inside the damper creates the resistance needed to control (dampen) the suspension movement. How much resistance (force) is developed for a given velocity is determined by internal valves that control the flow of oil. Based on the pressure (velocity), the valves open or close and create a given damper force for each bump or vehicle body roll. Most of the existing dampers do this; but the FSD damper (developed by Koni) adds a second valve system that responds to the damper's operating frequency, allowing ride and handling to be tuned more independently than with conventional dampers. The damper force created by the normal dampers only depend on the input velocity, while the damper force created by the FSD damper also depends on the frequency of this input velocity.

**Advantages:**
- Less compromise between comfort and handling
- Improved ride comfort and/or road holding
- Can be integrated in conventional damper layouts,
- Low cost solution, no electronics, sensors or other components needed

**Disadvantages:**
- New unproved development

**Twin-tube CDC Design**

Continuous Damping Control (CDC) calculates all forces required from the shock absorbers as a function of the vehicle load and the nature of the road. It does this for braking and accelerating, on bends or on gradients. A control unit, the system's brain, electro magnetically regulates a proportional value which determines the flow rate of the oil in the damper. If, for example, the valve is under a higher voltage, the opening in the damper is made smaller. The oil flows slower and the damping is "harder". The system calculates and regulates the damping force every 25 milliseconds and thus constantly reads the entire driving situation. Height and pressure sensors in the vehicle supply the data for the calculations. The CAN (controller area network) bus also supplies the damper control system with information: road speed, retardation or acceleration signals, lateral acceleration and wheel speed are all included in the precise calculations. In just fractions of a second the CDC adjusts a basic damping applying in normal conditions to the new circumstances.

**Mono-tube design**

These are high-pressure gas dampers with only one tube, the pressure tube. Inside the pressure tube there are two pistons: a dividing piston and a working piston. The working piston and rod are very similar to the twin tube shock design. A mono-tube shock absorber can be mounted upside down or right side up it will work either way. In addition to its mounting flexibility, mono-tube damper are a significant component, along with the spring, in supporting vehicle weight. The mono-tube shock absorber does not have a base valve. Instead, all of the control during compression and extension takes place at the piston. The pressure tube of the
mono-tube design is larger than a twin tube design to accommodate for dead length. A free-floating dividing piston travels in the lower end of the pressure tube, separating the gas charge and the oil. The area below the dividing piston is high pressurized with nitrogen gas. This high gas pressure helps to support some of the vehicle's weight. The oil is located in the area above the dividing piston. During operation, the dividing piston moves up and down as the piston rod moves in and out of the shock absorber, keeping the pressure tube full all times.

Advantages:
- Can be mounted upside down, reducing the unsprung weight
- May run cooler since the working tube is exposed to the air

Disadvantages:
- Difficult to apply to passenger cars designed OE with twin tube designs.
- A dent in the pressure tube will destroy the unit

Magnetic fluid design
Dampers (shock absorbers) utilize a fluid flow system incorporating therein either a hydraulic fluid having a constant viscosity or a fluid having a changeable viscosity, e.g., magneto-rheological (MR) fluid. The use of MR fluid is advantageous in that the viscosity thereof can be controlled with the application of a magnetic field in order to adjust a damping force being exerted on the springs depending on a traveling condition. Particularly, MR fluid is a free-flowing liquid with a viscosity. Exposure to a magnetic field can transform the liquid into a near-solid in milliseconds; and with the removal of the magnetic field, the fluid can be returned to its liquid state just as quickly. The degree of change in the viscosity of the MR fluid is proportional to the magnitude of the applied magnetic field.

The Bose suspension system
The Bose system uses a linear electromagnetic motor (LEM) at each wheel in lieu of a conventional shock-and-spring setup. Amplifiers provide electricity to the motors in such a way that their power is regenerated with each compression of the system. The main benefit of the motors is that they are not limited by the inertia inherent in conventional fluid-based dampers. As a result, an LEM can extend and compress at a much greater speed, virtually eliminating all vibrations in the passenger cabin. The wheel's motion can be so finely controlled that the body of the car remains level regardless of what's happening at the wheel. The LEM can also counteract the body motion of the car while accelerating, braking and cornering, giving the driver a greater sense of control.
Conclusion

The twin-tube FSD design appears to give the most advantages regarding a tractor semi-trailer equipped with an IFS suspension. This twin-tube damper concept can replace the original twin-tube damper, so no further technical modifications are necessary. The main problem of the IFS's disappointing ride comfort is the difference in damper performance between the high frequency roll motion (driving over irregularities road put holes) en low frequency bump motions (driving over the waving road). The FSD damping force is frequency dependent, so with a good damper set-up the IFS ride comfort problems can be solved. To investigate the possibilities of the FSD damper a multi-body model block of de FSD damper will be developed. The original dampers of the IFS will be replaced by the FSD dampers, so the tractor semi-trailer can simulate with FSD dampers. The results will be compared with the original twin tube configuration.

6.2. FSD working principle

The internal FSD feature of the FSD dampers automatically adjusts the rate and shape of the force curve (when plotted on a graph), responding to the frequencies of the damper's movement. A secondary valve inside the FSD is used to create a damper force which is dependent on the body movement frequency; this secondary valve is engineered to provide large amounts of force to control the vehicle body movement at relatively low frequencies, while generating less force at the higher frequencies to improve the ride comfort.

In standard shock absorbers the main damping characteristic is defined by the oil flow going through the piston assembly (Figure 6-5 [1]). Combining it with the FSD feature [2], a special valve is added that controls an oil flow parallel to the one going through the piston rod [3]. This parallel oil flow is closed by the FSD feature; giving an negative exponential rise in damping force to the time that the piston is moving in one direction. This means the secondary valve, which is almost closed during rapid piston direction changes (high frequency), will slowly open when the piston direction changes becomes slower (low frequency).

Summarizing: the FSD feature is a hydraulic system that delays the build up of damping force, so the damper reacts progressive on the suspension travel velocity. A better combination of handling and comfort has been created, through use of a secondary frequency dependent damper valve. Because it is an integrated part of the hydraulic valve system inside the damper, no additional cables, sensors or any other electronic devices are needed to operate an FSD damper, which reduces the costs.
6.3. FSD multi-body model

The FSD damper could be a possible solution to improve the poor ride comfort of the tractor semi-trailer equipped with IFS. The short asymmetric wheel bumps, produce by the road pot holes, are particularly responsible for the degree of ride comfort. Those short suspension movements will become less severe using the FSD damper, which is softer for higher suspension travel frequencies.

To simulate the tractor semi-trailer model equipped with the FSD damper a multi-body model is made of the damper. The FSD damper model is part of a modified damper block. In the underlying damper block standard multi-body formulas like the in-product are used to compute the absolute suspension movement and velocity. The velocity is used as input to the FSD damper block (see Figure 6-6) which has a force output.

Because the FSD damper is time dependent the model must be able to measure the time while the damper is moving in one (bump or rebound) direction. This will be done using the sample and hold block, this block will hold at the point of time where the damper velocity was changing direction. Subtracting this point of time to the real simulation time will give the length of time of
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damper travel in one direction of a given moment. The measured time difference goes into the FSD correction factor lookup table, where the damping force correction factor is given as a function of travel time. Multiplying the FSD factor with the original non-linear damper parameters, by a given damper velocity and motion frequency, will result in the actual damping force. So when the FSD damper is moving in one direction for a longer time, the time of travel becomes higher which increases the damping force. In other words when the suspension frequency increase the damping force becomes smaller, so the forces leading into the tractor chassis will decrease, which benefit the ride comfort.

To show the working principle of the FSD damper some test cases are evaluated.

![Damper input velocity](image1)

![Normal twin-tube damper](image2)

![FSD damper](image3)

Figure 6-7: Damper force during different time steps

In Figure 6-7 the damping force is plotted against the time during three different time steps with all the same amplitude (damper velocity step) of 0.1 m/s shown by the first plot:

- First time step of 1 second is starting after 1 second
- Second time step of 0.5 second is starting after 3 seconds
- Third and last time step of 0.25 second is starting after 4 seconds
In the second plot, which presents the normal twin-tube damper, the time step input with amplitude of 0.1 m/s will directly produce a constant damper force. Looking at the third plot the stationary damper force will not immediately reached. Some delay is introduced before reaching the stationary damper force, which is a result of the FSD principle. When shorting the time steps (higher freq.) the stationary damper force will not be reached (see lower figure after 3 sec.).

In Figure 6-8 a sine sweep signal is used as velocity input. This signal produces a sine wave whose frequency varies linearly with a given time. A sine wave from 0.1 Hz to 10 Hz over a time interval of 60 sec. with a max velocity (amplitude) of 0.1 m/s is used to show the difference between a normal twin-tube damper and the FSD damper.

Figure 6-8 clearly shows the decrease of damper force of the FSD damper when increasing the input frequency, while the normal twin-tube damper shows a constant damper force during all frequencies.
A road profile does not exist of one sine function as shown in Figure 6-8. In reality a road profile consists of several sine waves together. There can be a lot of potholes in a road while driving over long road waves which results in low frequencies road irregularities combined with high frequencies. To look how the FSD response to several sine waves together the sine sweep used in Figure 6-7 and Figure 6-8 is summed by another high frequency low amplitude sine.

There is some fluctuation seen in Figure 6-9 looking at the resulting damper force when using a sine sweep with a second high frequency sine added. This fluctuation is caused by the fact that the maximum amplitude of the sine sweep will be higher or lower dependent on the height of the second sine. Finally the sum of the amplitudes of the input sine sweep and the second sine on a certain time determines the amplitude height or in other words the piston velocity, which is responsible for the acting damper force. The FSD principle of decreasing damper force by increased sine frequencies is still clearly shown.

Figure 6-7, Figure 6-8 and Figure 6-9 show that the FSD damper acts in a different more effective way to the different road disturbance. The high frequency asymmetric movements will by damped in a softer damper state which will benefit the ride comfort significantly.
6.4. FSD simulation results

Simulations have been run for both the rigid front axle suspension and IFS, with and without FSD dampers. The used FSD parameters are optimized for commercial truck use. Further on in this paragraph there will be a sensitivity analysis to the influence of those parameters to the overall ride comfort. Comparing the results of the simulations with the optimized FSD parameters gives the following results (Table 6-1):

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Leafspring</th>
<th>Leafspring with FSD</th>
<th>IFS</th>
<th>IFS with FSD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ride Comfort index [RCI]</td>
<td>0.51 m/s²</td>
<td>0.47 m/s²</td>
<td>0.76 m/s²</td>
<td>0.54 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG x-acc.</td>
<td>0.35 m/s²</td>
<td>0.34 m/s²</td>
<td>0.30 m/s²</td>
<td>0.26 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG y-acc.</td>
<td>0.48 m/s²</td>
<td>0.45 m/s²</td>
<td>0.92 m/s²</td>
<td>0.62 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG z-acc.</td>
<td>0.48 m/s²</td>
<td>0.49 m/s²</td>
<td>0.40 m/s²</td>
<td>0.40 m/s²</td>
</tr>
<tr>
<td>Dynamic wheel load</td>
<td>34123 N</td>
<td>33619 N</td>
<td>33941 N</td>
<td>33334 N</td>
</tr>
<tr>
<td>Suspension travel</td>
<td>4.4 mm</td>
<td>5.5 mm</td>
<td>8.6 mm</td>
<td>9.6 mm</td>
</tr>
</tbody>
</table>

Table 6-1: Comparing Leafspring suspension and IFS, with and without FSD dampers

Ride comfort improvement:

Rigid front axle normal dampers ⇒ Rigid front axle FSD dampers ≈ 8 % RCI improvement
IFS normal dampers ⇒ IFS FSD dampers ≈ 30% RCI improvement

Table 6-1 shows a large improvement of the ride comfort using the FSD dampers on the IFS suspension. The Ride Comfort Index of the IFS equipped with FSD dampers is now comparable to the rigid front axle suspension equipped with the normally used twin-tube dampers. The rigid front axle suspension with FSD dampers also gives some ride comfort improvement, however as shown the percentage of improvement is not as large as the improvement on the IFS. Because the large influence of the dampers to the IFS chassis roll motion (explained in chapter 5) the FSD dampers have large influence on the total ride comfort. The FSD realizes a soft ride when driving over high frequency asymmetric road put holes, while they behave firm when driving over symmetric low frequency road waves.

Figure 6-10 shows some interesting acceleration PSD’s plots of the rigid front axle suspension and the IFS comparing to the IFS equipped with the FSD damper. The front axle z-acceleration and roll of the IFS with FSD shows a high increase looking to the higher frequencies. The front axle can travel more because there is less interaction with the truck chassis, because the dampers react less intensive during higher frequencies. This improvement takes care of less force into the front chassis, which reduce the chassis torsion motion. So the cabin has to deal with less intense vibrating chassis, which improve the ride comfort significantly.
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Figure 6-10: z-acc and roll-acc of the frontaxle, chassis and cabin during tractor semi-trailer simulation with respectively the leafspring front suspension, IFS and IFS equipped with the FSD dampers
Sensitivity of the FSD starting point

In models simulated so far the FSD dampers use a lookup-table which starts from zero, this means there is no damping correction at time when the damping travelling time is zero. This means the FSD dampers have zero resistance during high frequency motions. Design such damper is technical impossible. To get a look on more realistic situations the FSD damping correction factor starting point of the look-up table is shifted to more realistic values as shown in Figure 6-11.

Figure 6-11: Look-up table variation in correction factor starting point from 0 to 0.5
(Shifting of the FSD exponential starting point over the y-axle)

The shifting of the damping correction factor from 0 to 0.2 at travelling time zero results in a deterioration of the ride comfort as shown in Table 6-2:

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Leafspring</th>
<th>IFS</th>
<th>IFS &amp; FSD Start at 0</th>
<th>IFS &amp; FSD Start at 0.2</th>
<th>IFS &amp; FSD Start at 0.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ride Comfort index [RCI]</td>
<td>0.51 m/s²</td>
<td>0.76 m/s²</td>
<td>0.54 m/s²</td>
<td>0.60 m/s²</td>
<td>0.66 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG x-acce.</td>
<td>0.35 m/s²</td>
<td>0.30 m/s²</td>
<td>0.26 m/s²</td>
<td>0.26 m/s²</td>
<td>0.27 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG y-acce.</td>
<td>0.48 m/s²</td>
<td>0.92 m/s²</td>
<td>0.62 m/s²</td>
<td>0.71 m/s²</td>
<td>0.79 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG z-acce.</td>
<td>0.48 m/s²</td>
<td>0.40 m/s²</td>
<td>0.40 m/s²</td>
<td>0.38 m/s²</td>
<td>0.37 m/s²</td>
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<tr>
<td>Dynamic wheel load</td>
<td>34123 N</td>
<td>33941 N</td>
<td>33334 N</td>
<td>33734 N</td>
<td>33836 N</td>
</tr>
<tr>
<td>Suspension travel</td>
<td>4.4 mm</td>
<td>8.6 mm</td>
<td>9.6 mm</td>
<td>9.0 mm</td>
<td>8.8 mm</td>
</tr>
</tbody>
</table>

Table 6-2: Comparing Leafspring suspension and IFS, with different y-axis starting points
Looking at the difference in the PSD's plots (see Figure 6-12) it is obvious the frontaxle introduces more forces into the chassis when the damping correction factor at time zero is increased.

Figure 6-12: z-acc and roll-acc of the frontaxle, chassis and cabin during tractor semi-trailer simulation with the IFS equipped with shifted y-axis values of the FSD look-up table
Sensitivity of the FSD ending time

There are many more parameters which influence the characteristics of the FSD damper. Investigating them all will be a study case by itself. This FSD investigation will be restricted to a sensitivity analyze by shifting the exponential FSD lookup table over the y-axis and x-axis. Results of shifting over the y-axis, which means increasing the damping correction factor at starting point time is zero, are shown in the previous paragraph.

Besides shifting the y-axe damping correction factor as done before, also the x-axe ending time (the time where the exponential function reach one) can be shifted. Varying this end point along the x-axe gives a change in the damper frequency sensitivity. When shifting up the ending time the FSD damper will show more frequency sensitivity, the time window of the exponential function will be longer. Reaching higher damping forces during one direction damper travel will take more time, which automatically means the damping forces will be lower during short damper movements. In Figure 6-13 this parameter variation is shown.

Figure 6-13: FSD Look-up table ending time variation form 0.19 to 1.50 seconds comparable to 25% - 200% of the normal used 100% reference point
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<table>
<thead>
<tr>
<th>Parameters</th>
<th>Leafspring</th>
<th>IFS</th>
<th>IFS &amp; FSD 50% x-axis</th>
<th>IFS &amp; FSD 100% x-axis</th>
<th>IFS &amp; FSD 150% x-axis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ride Comfort index [RCI]</td>
<td>0.51 m/s²</td>
<td>0.76 m/s²</td>
<td>0.59 m/s²</td>
<td>0.54 m/s²</td>
<td>0.53 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG x-acc.</td>
<td>0.35 m/s²</td>
<td>0.30 m/s²</td>
<td>0.27 m/s²</td>
<td>0.26 m/s²</td>
<td>0.26 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG y-acc.</td>
<td>0.48 m/s²</td>
<td>0.92 m/s²</td>
<td>0.71 m/s²</td>
<td>0.62 m/s²</td>
<td>0.59 m/s²</td>
</tr>
<tr>
<td>RMS Cabin CG z-acc.</td>
<td>0.48 m/s²</td>
<td>0.40 m/s²</td>
<td>0.38 m/s²</td>
<td>0.40 m/s²</td>
<td>0.42 m/s²</td>
</tr>
<tr>
<td>Dynamic wheel load</td>
<td>34123 N</td>
<td>33941 N</td>
<td>33695 N</td>
<td>33334 N</td>
<td>33495 N</td>
</tr>
<tr>
<td>Suspension travel</td>
<td>4.4 mm</td>
<td>8.6 mm</td>
<td>8.8 mm</td>
<td>9.6 mm</td>
<td>10.5 mm</td>
</tr>
</tbody>
</table>

Table 6-3: Comparing Leafspring suspension and IFS, with different FSD ending times

In Figure 6-14 again the PSD plots of the front axle, chassis and cabin z-acceleration and roll motion are shown, but now with different FSD look-up table x-axis parameters. The ending time is shifted to the left to 50% and to the right to 150% of the reference. As reference the simulation parameters used in the beginning of this paragraph are taken. Looking at Figure 6-14 it can be concluded that shifting the exponential function ending time to the right (to later ending times) the PSD motion of the front axle increase significantly. The front axle can travel more during high frequency motions so there is less interaction with the chassis which benefits the ride comfort. A question is what happens with the road holding when the front tyres can move with reduce damping forces, which must take into account when designing a FSD damper for a heavy duty truck.
Figure 6-14: z-acc and roll-acc of the frontaxle, chassis and cabin during tractor semi-trailer simulation with the IFS equipped with shifted x-axis values of the FSD look-up table.
Conclusion

A significant ride comfort improvement will be achieved when equipping the IFS with FSD dampers. Also for the original rigid front axle suspension FSD dampers can contribute to ride comfort improvement.

In Figure 6-15 and Figure 6-16 the starting point and ending time are plotted against the corresponding damping correction factor and end time, the time where the exponential function reach one. This gives an overview of the ride comfort sensitivity of those two variables.

Figure 6-15: FSD sensitivity results of the RCI driver’s seat vs. FSD variable damping the correction factor
It is necessary to keep the beginning damping force (in the beginning of the suspension travel at time zero) of the FSD as low as possible for the best working of the FSD. The damping correction factor which is responsible for the damping force is almost linear depending on the ride comfort (see Figure 6-15). The height of the beginning damping force is depending on the FSD design. Figure 6-16 tells us the ride comfort is exponential depending on the FSD ending time. A good point of departure is 1 second; however this value must be adjusted to the drivability. Higher ending times influence the drivability unfavorable.

Using the FSD damper will result in high ride comfort improvements. The exact height of the improvement is strongly depending on the damper parameters. By studying and analyzing those parameters the best FSD damper setup can release.
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7 Conclusions and recommendations

7.1 Conclusions

An independent front suspension is evaluated using a tractor semi-trailer model and compared with a rigid frontaxle suspension. This is realized by building an exchangeable frontaxle model block which fits into the existing tractor semi-trailer multi-body model. The main simulations consist of straight line driving over a measured road profile with a velocity of 85 km/h.

The expected ride comfort improvement originating from an independent front suspension appears to be absent. In contrast with the expectations, the ride comfort appears to be worse than the original rigid frontaxle suspension.

After investigation why the ride comfort aggravates, the chassis beam construction of the commercial vehicle appears to be the biggest factor. In contrast with a passenger car which is built on a very stiff integral-frame construction, commercial vehicles use a weak framework chassis. The large spring and damper track width of the IFS (left upright – right upright) compared to the small track width of the rigid frontaxle suspension (left chassis longitudinal runner – right chassis longitudinal runner) are responsible for bigger forces to the chassis of the IFS. This results in more chassis roll motion, which feel reinforced by the driver due to opposite reaction of the cabin suspension. The high frequency road disturbances created by put-holes are responsible for asymmetric suspension travel, which creates the straight line driving roll motion.

Designing a stiffer tractor chassis can solve this problem, but after investigation of the torsion stiffness influence to the ride comfort, it appears the torsion stiffness must be 25 times as high as the regularly torsion stiffness. Since raising the torsion stiffness of maximal 3 times the regular one is feasible, reinforcement of the chassis is not a practical solution.

To improve the ride comfort of a truck with IFS the most important suspension parameters are investigated. After sensitivity analysis of those parts, the dampers appear to have large impact on the ride comfort. Improving the damping characteristics is the most obvious opportunity for correcting the ride comfort of the IFS. After reviewing all kind of advance shock absorber designs the best design is chosen, this is the FSD damper. The FSD damper is build into the IFS model block. Using FSD dampers in the IFS tractor semi-trailer configuration gives a 30% ride comfort improvement. Although FSD dampers also lead to a ride comfort improvement for the rigid frontaxle suspension, the improvement is smaller, namely 8%. The FSD dampers create most ride comfort improvement in the high frequency range of the IFS.
It can conclude that building an independent suspension under a weak truck chassis will not give the benefits like building an independent suspension under a stiff integral passenger car frame. A truck builder has taken the different advantages and disadvantages into account when using an independent front suspension on a truck.

7.2. Recommendations

The following recommendations can give for improving the investigation of the IFS:

- Use a more advanced chassis multi-body model with a finite-element analysis, to make better and more realistic simulations
- Design a subframe construction for better suspension force distribution to the tractor chassis
- Improve the design of the modeled IFS, so more benefits of a double A-arm suspension will be used.
- Look at the opportunities designing an integral commercial vehicle frame like used in busses.
- Adding an extra spring damper mass body which acts as air suspended driver seat for a more realistic ride comfort simulation.
- Fine tune the FSD damper settings especially the used lock-up table.
- Look to the consequences of the FSD damper to the drivability and stability of the tractor semi-trailer model.
- Steering maneuvers must simulate to take the drivability into account.
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8.1. IFS design examples

Some design examples of an independent front suspension for heavy vehicles:

Figure 8-1: IFS with rotational uprights, spring and damper mounted on the upper and lower A-arms (Source: IFS1260, Reyco Granning suspensions)

Figure 8-2: IFS with dampers mounted to the lower A-arms (Source: IFS1660, Reyco Granning suspensions)

Figure 8-3: IFS with rotational uprights, spring and damper mounted on the lower A-arms (Source: TAK-4 IFS, Pierce Manufacturing Inc.)
8.2. Road data

In practice road profiles are random signals. Figure 8-4 presents a 1200 m long road profile, measured with a sample interval of 0.05 m. In the upper plot the complete road profile and in the lower plot a 50 m section of the road profile are plotted. Observed can be that low frequency unevenness has large amplitudes, whereas high frequency unevenness has small amplitudes.

![Figure 8-4: Measured road profile](image)

It is common to represent a road profile by its displacement power spectral density (PSD). In Figure 8-5 the power spectral density of the measured road profile is plotted. In addition, the ISO road class limits A-H are shown [ISO 8608:1995(E)].
The spatial frequency $n$ is defined by the relation:

$$n = \frac{1}{\lambda} = \frac{f}{V}$$

Where $\lambda$ is the road surface wavelength, $f$ the frequency and $V$ the vehicle forward velocity.

Measurements on various road types have revealed that a slope about –2 on a log-log scale appears for many road surfaces. In addition, it has been found that different road surface types (smooth asphalt, Belgian blocks, etc.) will mainly result in a vertical shift of the PSD. Therefore, in [ISO 8608:1995(E)] the following expression is proposed to represent a road profile:

$$S_z(n) = S_z(n_0) \left( \frac{n}{n_0} \right)^2 \quad \text{with} \quad n_0 = 0.1 \text{m}^{-1}$$
Closely related to this relation are the road classes A-H. Table 8-1 presents the $S_{nx} (n)$ values that belong to the class limits. For simulation purpose, the mean values may be used. Finally, notice that this spatial frequency domain PSD only considers geometrical road surface data.

<table>
<thead>
<tr>
<th>Road class</th>
<th>Upper limit $[10^{-6} \text{ m}^3]$</th>
<th>Geometric mean $[10^{-6} \text{ m}^3]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (very good)</td>
<td>$2^5$</td>
<td>$2^4$</td>
</tr>
<tr>
<td>B (good)</td>
<td>$2^7$</td>
<td>$2^6$</td>
</tr>
<tr>
<td>C (average)</td>
<td>$2^9$</td>
<td>$2^8$</td>
</tr>
<tr>
<td>D (poor)</td>
<td>$2^{11}$</td>
<td>$2^{10}$</td>
</tr>
<tr>
<td>E (very poor)</td>
<td>$2^{13}$</td>
<td>$2^{12}$</td>
</tr>
<tr>
<td>F</td>
<td>$2^{15}$</td>
<td>$2^{14}$</td>
</tr>
<tr>
<td>G</td>
<td>$2^{17}$</td>
<td>$2^{16}$</td>
</tr>
<tr>
<td>H</td>
<td>-</td>
<td>$2^{18}$</td>
</tr>
</tbody>
</table>

Table 8-1: Classification of road roughness according to ISO 8608:1995(E)
8.3. Tyre modelling

In a multi-body modelling environment the tyre can be considered as a force element. In the direction normal to the road the tyre behaves as a spring/damper and for motions perpendicular to the road plane the tyre develops reaction forces as a result of the relative (sliding) motion with respect to the road surface. Unfortunately these force relations are quite complex and highly non-linear. To model the tyre forces two approaches can be followed:

- physical tyre modelling
  Based on the physical properties of the tyre (material, dimensions,...) a detailed mechanical model is created. In order to have some accuracy one rapidly has to resort to a coarse FEM model having already many degrees of freedom; simple brush type tyre models having analytical solutions will be insufficient.

- (semi-) empirical tyre modelling
  In this approach the measurements on the rolling tyre are the basis for the model. An obvious example is to store measurement data in a look-up table, which is then evaluated later during the simulation. Tyre models in this class would typically use e.g. spline interpolation or special mathematical formulae (like the Magic Formula).

Semi-empirical tyre modelling has the advantage that the resulting tyre model tends to be accurate and fast, but it relies on tyre measurements. Physical tyre models may have better predictive qualities when no measurement data is available, but will be slower and ultimately don’t achieve the same accuracy as a semi-empirical tyre model or at high computational costs.

It is sufficient to say that TNO Automotive has supported the automotive industry for more than 10 years in applying the Magic Formula. The activities range from doing on the road tyre measurements, parameter identification and providing tyre models for different multi-body simulation packages, as shown in Figure 8-6.

Figure 8-6: Capturing tyre behaviour from tyre testing to Multi-body simulation
The Magic Formula is used to describe the non-linear steady-state representation of the tyre characteristics. Next to that the dynamic behaviour of the tyre has to be included. Taking into account the tyre relaxation behaviour is a first extension to cover the tyre dynamics up to 8 Hz. At higher frequencies the wheel and tyre cannot be modelled as a single rigid body anymore. To include tyre dynamics up to 60-80 Hz it appears to be sufficient to describe the motions of the tyre belt and rim using two separate bodies, as shown in Figure 8-7. Furthermore an enveloping model is necessary to cope with short wavelength road obstacles [I.J.M. Besselink, Vehicle Dynamics analysis using SimMechanics and TNO Delft-Tyre].

Within the MATLAB, Simulink and SimMechanics environment these tyre models are supported in various ways. First a command line function exists to evaluate the Magic Formula (e.g. for the purpose of plotting of tyre characteristics). Next two Simulink blocks to include the Magic Formula in a vehicle model are available. And finally there’s an easy to use block to include the wheel/tyre assembly in a SimMechanics model, which is used in the tractor semi-trailer SimMechanics model (Figure 8-8).
8.4 Ride comfort weighting functions

This SimMechanic Block makes use of measured road data file and three different tyre property files namely:

1. Tractor front axle (steering axle): Goodyear 315/80 R22.5
2. Tractor rear axle (driven axle): Bridgestone 315/80 R22.5
3. Trailer axles: Goodyear 385/65 R22.5

With this future the tractor semi-trailer model will be simulated in a real environment, diving about a measured road profile that creates tyre force to the measured tyres, which implements chassis forces using the Magic Formula with the Delft-tyre block.
8.4. Ride comfort weighting functions

The sensitivity to horizontal vibrations is somewhat different from that of vertical. The most remarkable difference is that the region of maximum sensitivity occurs in the 0.5 to 2 Hz range. In force/aft direction, this sensitivity is generally recognized to result from the fore/aft resonance of the upper torso. So as mentioned before, ride comfort is assessed in the 0.5 to 80 Hz range. Above 80 Hz, the human body is not very sensitive to vibrations. In the 0.1 to 0.5 Hz range, vibrations may cause motion sickness. To account for the human body sensitivity for specific frequencies, weighting functions $H_{\text{iso}}(f)$ are applied to the vertical, longitudinal and lateral acceleration signals. As shown in figure 4-3, weighting functions are defined for the vertical and horizontal directions. In addition, a weighting function is defined for assessing motion sickness (only vertical direction).

![Graph showing ISO 2631-1, 1997 weighting functions](image)

By applying the weighting function for vertical direction, the frequency-weighted vertical acceleration of the sprung mass $\ddot{z}_{s,\text{iso}}(f)$ is obtained as follows:

$$\ddot{z}_{s,\text{iso}}(f) = H_{\text{iso},z}(f) \cdot \ddot{z}_s(f)$$

In terms of power spectral densities, this equation reads:

$$S_{z_{s,\text{iso}}}(f) = |H_{\text{iso},z}(f)|^2 \cdot S_{z_s}(f)$$
For vibrations in more than one direction, the vibration total value of weighted RMS acceleration $a_v$, determined from vibration in orthogonal coordinates is calculated as follows:

$$a_v = \sqrt{k_x^2 a_{wx}^2 + k_y^2 a_{wy}^2 + k_z^2 a_{wz}^2}$$

Where:
- $a_{wx}$, $a_{wy}$ and $a_{wz}$ are the weighted RMS accelerations with respect to the orthogonal axes $x$ (longitudinal), $y$ (lateral), $z$ (vertical), respectively;
- $k_x$, $k_y$ and $k_z$ are multiplying factors.

For assessing comfort of seated persons, what is the case in the DAF tractor semi-trailer ride comfort simulation, the factors $k_x$, $k_y$ and $k_z$ equal 1. According to [ISO 2631-1, 1997] it is recommended to use the vibration total value $a_v$ for assessing comfort. Note that the weighted acceleration values can be obtained by calculating the areas under the corresponding weighted power density spectra’s. Acceptable values of vibration magnitude for comfort depend on many factors. Therefore a limit is not defined in (ISO 2631-1, 1997, ref. [14]). Nevertheless, the values in Table 4-1 give approximate indications of likely reactions to various magnitudes of overall vibration total values in road transport.

<table>
<thead>
<tr>
<th>Vibration total value:</th>
<th>Likely reaction:</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 0.315 m/s²</td>
<td>Not uncomfortable</td>
</tr>
<tr>
<td>0.315 m/s² to 0.63 m/s²</td>
<td>A little uncomfortable</td>
</tr>
<tr>
<td>0.5 m/s² to 1 m/s²</td>
<td>Fairly uncomfortable</td>
</tr>
<tr>
<td>0.8 m/s² to 1.6 m/s²</td>
<td>Uncomfortable</td>
</tr>
<tr>
<td>1.25 m/s² to 2.5 m/s²</td>
<td>Very uncomfortable</td>
</tr>
<tr>
<td>&gt; 2 m/s²</td>
<td>Extremely uncomfortable</td>
</tr>
</tbody>
</table>

Table 8-2: Vibration total values reactions

The vibration total value is also called the ride comfort index RCI.